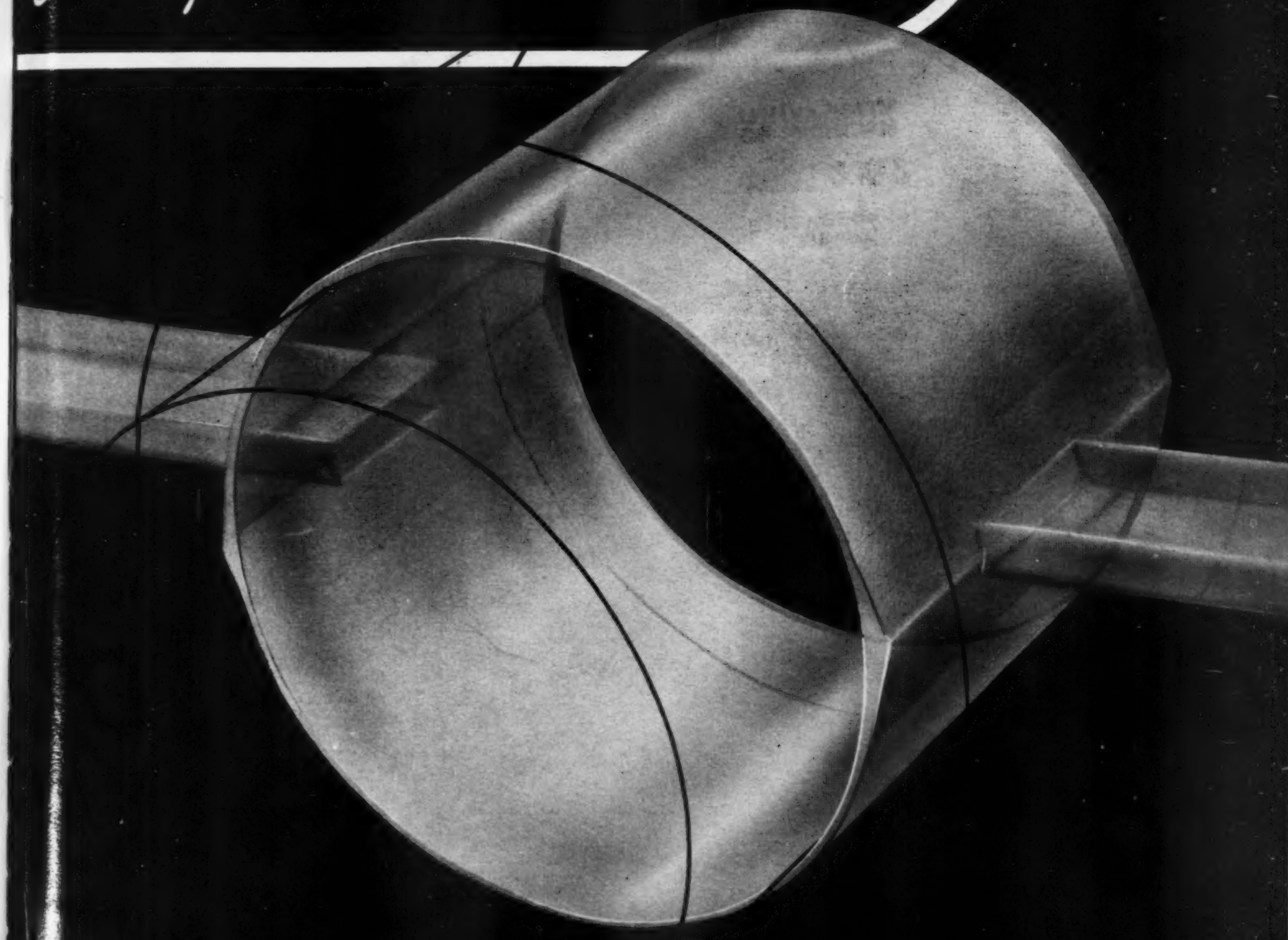
MACHINE DESIGN

August 1953



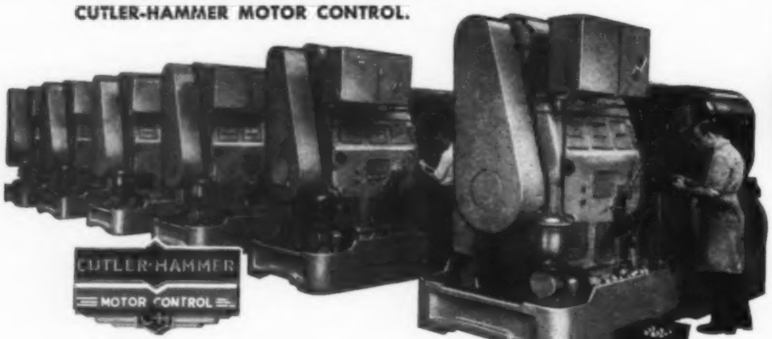
JOURNAL BEARING LUBRICATION

Contents, Page 3

Choice of the Leaders

CUTLER-HAMMER MOTOR CONTROL C-H

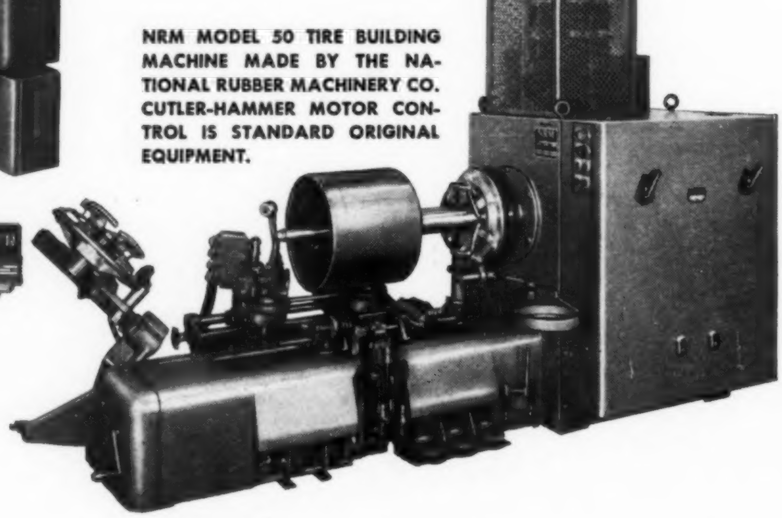
The Mark of Better Machines



BATTERY OF ACME-GRIDLEY BAR AUTOMATICS MADE BY THE NATIONAL ACME COMPANY. THESE MACHINES SHOWN ON TEST FLOOR ARE EQUIPPED FULLY WITH CUTLER-HAMMER MOTOR CONTROL.

PACIFIC PORTABLE TRUCK LOADING HOIST MADE BY HOIST DIVISION, RUSH-LIGHT'S, INC. CUTLER-HAMMER CONTROL IS STANDARD ORIGINAL EQUIPMENT.

CINCINNATI 20" RAPID TRAVERSE SHAPER MADE BY THE CINCINNATI SHAPER CO. CUTLER-HAMMER MOTOR CONTROL IS STANDARD ORIGINAL EQUIPMENT.



NRM MODEL 50 TIRE BUILDING MACHINE MADE BY THE NATIONAL RUBBER MACHINERY CO. CUTLER-HAMMER MOTOR CONTROL IS STANDARD ORIGINAL EQUIPMENT.

Synonyms for Satisfaction

When products gain such widespread approval that they are accepted as the standard for comparison, it can be truly said that the names by which they are known are synonyms for satisfaction. That position of genuine leadership is never a mere happenstance. It can stem only from superior product performance. And this outstanding performance is invariably the result of strong and rigidly enforced policies governing product quality. Without such policies, leadership can never be won. Without them it can never be upheld. Probably nothing shows both the force and effect of these policies more clearly than the care with which leading machinery builders select the electrical equipment which becomes so much a part of their products' performance. CUTLER-HAMMER, Inc., 1310 St. Paul Avenue, Milwaukee 1, Wisconsin. *Associate:* Canadian Cutler-Hammer, Ltd., Toronto, Ontario.

THE PROFESSIONAL JOURNAL
FOR ENGINEERS AND DESIGNERS

MACHINE DESIGN

August, 1953
Vol. 25—No. 8

REGULAR DEPARTMENTS

USE POSTAGE-FREE CARDS FOR:

Over the Board

Donates Machine Design Check

We experienced a warm glow upon reading a news story in the *Lynn* (*Mass.*) *Item* of June 25. The story tells how the newspaper's Worcester Tornado Relief Fund was boosted by a check presented by Isaac F. Kinnard, manager of engineering, Meter and Instruments Dept. of G.E.'s West Lynn Works, going on to say:

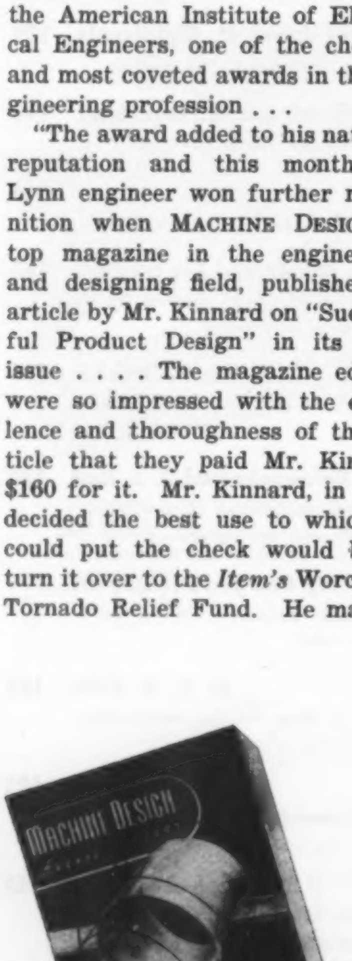
"Behind this gift is a heartwarming story of generosity and public spirit on Mr. Kinnard's part. It will be recalled that the gifted G.E. executive was recently presented the Lamme Gold Medal of the American Institute of Electrical Engineers, one of the choicest and most coveted awards in the engineering profession . . .

"The award added to his national reputation and this month the Lynn engineer won further recognition when MACHINE DESIGN, a top magazine in the engineering and designing field, published an article by Mr. Kinnard on "Successful Product Design" in its June issue . . . . The magazine editors were so impressed with the excellence and thoroughness of the article that they paid Mr. Kinnard $160 for it. Mr. Kinnard, in turn, decided the best use to which he could put the check would be to turn it over to the *Item's* Worcester Tornado Relief Fund. He made a

special trip to the *Item* office with Carl Svenson [seen holding copy of MACHINE DESIGN], who has assisted him in working out the philosophy expounded in the article, endorsed the check to the *Item* fund and handed it personally to Charles E. Gallagher, managing editor [seen standing at right of picture]."

This Month's Cover

Ever stop to think how the oil film in a bearing would look if the bearing were removed before the film fell apart? Through the magic of artist George Farnsworth's brush we are able this month to reveal such a film in its true form and color. Only the ghosts of the journal and bearing remain. The cover picture highlights the theme of Wilcock and Rosenblatt's article on "Nomographic Method for Sleeve Bearing Design" which begins on Page 143.



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MACHINE DESIGN
AUGUST 1953

One in the Hand or Two in the Bush?

IN THE CURRENT mad scramble for engineering manpower, starting rates for engineers just out of college now average an all-time high of $375 per month. While such recognition of the value of engineering talent is gratifying, what does it do to the salary structure of engineering departments that include men of experience who were hired at starting rates of $125?

Karl B. McEachron recently told the American Society for Engineering Education that proper adjustment in salaries of experienced men to maintain ratios that are fair and just will soon become impossible if the trend continues. Using the criterion that the present salary of a professional man with fifteen or twenty years experience should be at least double the present starting rate for new graduates, coupled with the fact that starting rates themselves have trebled in fifteen years, he draws a salary curve for such an individual which appears, as he puts it, almost asymptotic—a sixty-four dollar word for reaching toward infinity.

If engineering organizations are to function in a normal manner and continue attracting competent men there must be a reasonable spread between salaries of experienced and inexperienced engineers, as well as considerable latitude for granting merit increases. The present trend spells trouble in these respects unless management comes to grips with the situation.

Engineering executives should take a long hard look at their existing setup and attempt to answer realistically such questions as, "Shall we hire ten new men or can we do better by hiring only five and raising the salaries of our experienced men?" Such an attitude, if universally adopted, could go a long way toward easing the apparent manpower shortage, and keeping starting rates within reasonable bounds.

Current publicity on the alleged shortage of engineering manpower overemphasizes quantities of engineers and underemphasizes quality. In the race for quantity let's not overlook the quality that may be already available in seasoned engineering employees. A dollar spent developing that quality may be worth two spent acquiring quantity. He was himself a wise old bird who said that a bird in the hand is worth two in the bush.

Colin Carmichael
EDITOR

	CLASS	0 BUT NOT .126"	.126" BUT NOT 1"	1" BUT NOT 2"	2" BUT NOT 3"	3" BUT NOT 4"	4" BUT NOT 5"	5" BUT NOT 6"
	6	6	12	18	24	30	36	42
BLACK	5	5	10	15	20	25	30	35
AMBER	4	4	8	12	16	20	24	28
BLUE	3	3	6	9	12	15	18	21
GREEN	2	2	4	6	8	10	12	14
RED	1	1	2	3	4	5	6	7

UNIT (ten thousandths)

Fig. 1—Chart outlining the basic Pilot +2 system for holes. To find the class, add 2 to the inch size of the hole (ignoring fractions) and divide into the limit. For tolerance, add 2 and multiply by the class number

Based on actual practice, this proposed new system offers many unique engineering advantages, reconciling the inherent opposition to present complex systems and incompatibility

SIMPLIFIED LIMIT

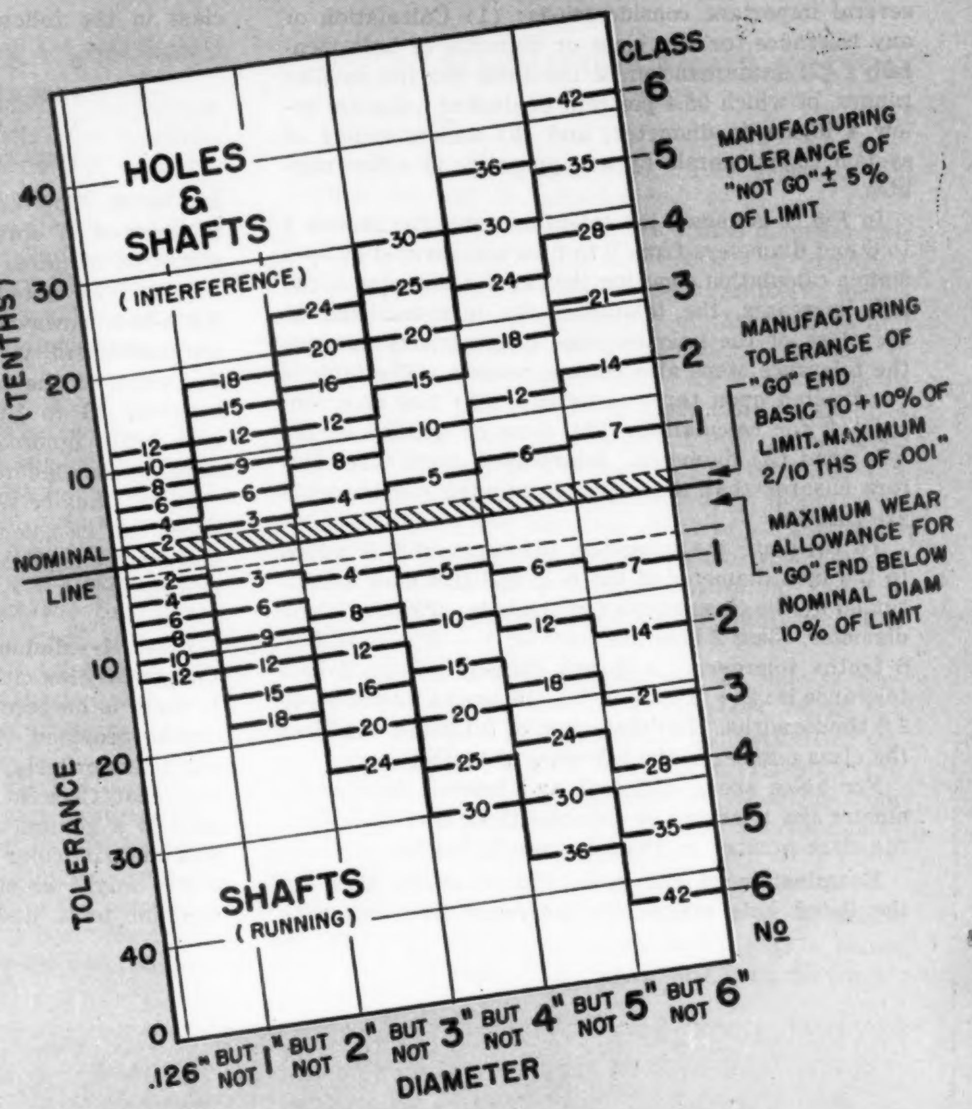
By F. W. M. Lee, *Managing Director, Pilot Plug Gauge Co. Ltd., Coventry, England*

IT HAS been apparent for a long time that a comprehensive fit and limit system, embracing the whole of the engineering field, would be both desirable and invaluable. But, it has been generally felt that if a system could be evolved, and in view of the many aspects it would have to embrace, it would be both unwieldy and complicated. However, such has not been the case and in this article a recommended simplified limit system will be proposed.

In a short article, space will not allow reference to the research that was undertaken in arriving at the final design, except to say that many systems were analyzed and superimposed upon one another graphically to see if they had any common features. Because there were none, it became abundantly clear that any proposed system had to be based upon a new conception in keeping with modern requirements.

Practical Magnitudes of Tolerance: In order to ascertain the tolerances generally employed in engineering today, nearly 10,000 limits on gages of recent manufacture were analyzed. From this analysis, a wealth of information concerning actual limits was obtained and, with these tolerances in mind, a new system—called the Pilot +2 system—constructed upon a number of practical ideals was designed. These ideals are not necessarily placed in order of importance, but are so arranged that they will assist in

Fig. 2—Complete layout of the new system as applied to shafts and holes with running and interference fits



SYSTEM

the logical development of the system.

Ideal 1—A Unilateral System: The unilateral system is based upon a standard hole and the diameter of the shaft varied to give the requisite fit, as opposed to the bilateral system in which the shaft is made to a standard and the hole varied to provide the fit. For engineering reasons, it is easier to vary the diameter of a shaft than it is to alter the size of a hole and this very practical fact indicates that an ideal system must be unilateral. In support of this conclusion, the dominating tolerance systems are unilateral.

The Pilot +2 system is therefore unilateral, by which is meant the smallest diameter of the hole is never less than the basic diameter and the tolerance, or the amount of error which will be tolerated, is always in a plus direction. That is, the largest diameter to which the hole may be made is greater than the basic size of the hole by the tolerance.

Ideal 2—Suitable Tolerances: As mentioned, nearly 10,000 gage limits were analyzed and details of this task cannot possibly be shown here. Only inspection of the chart which follows will indicate whether adequate tolerances have been provided for any particular product. It is felt that this ideal is attained.

Ideal 3—Simplicity: A simple system must permit

several important considerations: (1) Calculation of any tolerance for any class or diameter of hole mentally; (2) memorization of the table for the smaller ranges, of which 98.4 per cent of limited holes are below 4 inches in diameter; and (3) understanding of system fundamentals by an apprentice in a few minutes.

In *Fig.* 1 is shown the tolerance table for classes 1 to 6 and diameters from 0 to 6 inches, arrived at by a simple calculation ensuring that as the diameter of the hole increases, the tolerance also increases and as the class of the hole becomes progressively coarser, the tolerance steps also become coarser. The table is constructed upon three essentials: (1) Use of a constant 2 for calculation; (2) class or quality of the hole; and (3) diameter. Interplay of these three factors ensures that the above mentioned requirements are fulfilled.

To calculate the tolerance, the constant 2 is added to the inch diameter of the hole and this sum is multiplied by the class number of the hole. Thus, a 1-inch diameter, Class 2 hole tolerance is: $1 + 2 = 3 \times 2 =$ 6 tenths tolerance. A 2-inch diameter, Class 4 hole tolerance is: $2 + 2 = 4 \times 4 = 16$ tenths tolerance, or 1.6 thousandths. For diameters of 0.125-inch and less the class number is the *tolerance in tenths.*

For holes above .125 inch and below 1 inch in diameter the tolerance is obtained thus: $0 + 2 = 2 \times$ the class number $=$ the tolerance in tenths.

Examination of the table, *Fig.* 1, shows that for the listed hole ranges the tolerance increases with class in the following graduated steps:

0 -0.125 in steps of 1 tenth.
0.126-0.999 in steps of 2 tenths.
1 -1.999 in steps of 3 tenths.
1.999-2.999 in steps of 4 tenths.

and so on. The table can therefore be memorized, and the method of obtaining the tolerance explained in a matter of minutes.

It will be noted, apart from the modification of 0.125-inch downwards—incorporated for the use of the instrument, watch, clock, and similar areas—that the value of the tolerance changes at nearly 1 inch intervals (1 to 1.999 inches) fractional sizes of an inch being ignored. The soundness of this method need not be elaborated here.

It will thus be seen that the Pilot +2 system tolerances can be calculated mentally, can be memorized, and simply explained to an apprentice in a matter of minutes.

Ideal 4—Minimum Total Tolerance Range: Although only six classes of tolerances are shown in *Fig.* 1, there is no theoretical limitation to the number of classes provided. Industry, however, has already decided this criteria, for it has been found by investigation that Classes 3, 4 and 5 are the most popular. This is a general assertion and must be modified to suit the particular product. In one case, Classes 2, 3, and 4 might be stipulated as suitable for design of machine tools, and in another with less exacting re-

Fig. 3—American tolerance table constructed with gage catalog data compared to the basic Pilot system to show close similarity of actual practices

AMERICAN					
ABOVE	.029	.825	1.510	2.510	4.510
TO AND INCLUD'G	.829	1.510	2.510	4.510	6.510
Z	10	12	16	20	25
Y	7	9	12	15	19
X	4	6	8	10	13
XX	2	3	4	5	6.5

UNITS (ten thousandths)

PILOT + 2					
CLASS	.126" BUT NOT 1"	1" BUT NOT 2"	2" BUT NOT 3"	3" BUT NOT 4"	4" BUT NOT 5"
4	8	12	16	20	24
3	6	9	12	15	18
2	4	6	8	10	12
1	2	3	4	5	6

UNITS (ten thousandths)

quirements the coarser tolerances of 3, 4, and 5 could be used.

The desirable range must be decided by the individual firm concerned. It might be suggested, however, that in specifying, for instance, Classes 3, 4, and 5 only may be used, tighter tolerance being permitted only by special permission from the Design and Production Departments. In this way, limits which are unnecessarily tight and have no bearing upon design or production requirements will be avoided and prevented from being perpetuated and increasing production costs.

Ideal 5—Tolerances for Shafts: The complete layout of *Fig.* 2 shows the system as applied to shafts and holes. It will be noted that class tolerance lines exactly similar to the hole tolerance lines *above* the basic line are provided *below* the basic line. These shaft lines are calculated in precisely the same manner as the hole tolerances, so that the tolerances *Fig.* 1 could be literally turned upside down and serve two purposes. The question of shaft tolerances will not be pursued here, as a full explanation with examples, will be given in the following discussion.

Ideal 6—Mental Picture of Fits: Here it would be well to recall that with a unilateral system, the smallest diameter of the hole—subject to qualification later—is never less than the basic diameter. In any fit, the smallest diameter of the hole is the dominating dimension, so it is the nominal or basic diameter of the hole which must be kept in mind when deciding a fit. To make this clear, three types of well defined fits will be considered.

RUNNING FIT: It is assumed a 1 inch diameter hole has been produced to its smallest size of exactly 1 inch in diameter, and a shaft is required to run in this hole with a minimum clearance of three tenths and a maximum clearance of six tenths. Reference to the class lines below the basic line for 1 inch diameter indicates that this particular shaft can be described by the figures 1 and 2, of which 1 is 3 tenths below the nominal line, and 2 is 6 tenths. In the Pilot + 2 system, the nominal line is indicated by means of an oblique stroke / and since the shaft of a running fit is less in diameter than the smallest or nominal size of the hole, the class number controlling the size of the shaft would be placed to the right, or minus side, thus, /12.

Because the largest diameter of the hole is controlled by the class number, or in other words the tolerance, the hole is then perfectly described in the case of a Class 1 hole by 10/. The 0 or zero is the nominal size irrespective of class or diameter, from which it follows that only one number, the class number, is required to describe the hole, since the other number is always zero and can be ignored.

The running fit can now be written 1/12, of which the first 1 is the Class 1 hole, and the 12 a shaft which is a minimum of 3 tenths, and a maximum of

Fig. 4—Equivalent chart of American and Pilot tolerance systems. Exact equivalents exist in many cases and slight differences are of little practical importance except in three cases with XX grade. Minus sign indicates smaller Pilot tolerance

DIAMETER STEPS PILOT +2	DIAMETER STEPS USA	USA XX TOL	PILOT+2 TOL	PILOT+2 CLASS	DIFFERENCE	USA X TOL	PILOT+2 TOL	PILOT+2 CLASS	DIFFERENCE	USA Y TOL	PILOT+2 TOL	PILOT+2 CLASS	DIFFERENCE	USA Z TOL	PILOT+2 TOL	PILOT+2 CLASS	DIFFERENCE
	.029																
.125		2	2	2	0	4	4	4	0	7	7	7	0	10	10	10	0
	.825	2	2	1	0	4	4	2	0	7	6	3	-1	10	10	5	0
.999		3	2	1	-1	6	6	3	0	9	8	4	-1	12	12	6	0
	1.510	3	3	1	0	6	6	2	0	9	9	3	0	12	12	4	0
1.99		4	3	1	-1	8	9	3	+1	12	12	4	0	16	15	5	-1
	2.510	4	4	1	0	8	8	2	0	12	12	3	0	16	16	4	0
2.99		5	4	1	-1	10	8	2	-2	15	16	4	+1	20	20	5	0
3.99		5	5	1	0	10	10	2	0	15	15	3	0	20	20	4	0

MINUS SIGN INDICATES PILOT + 2 TOLERANCE IS SO MANY TENTHS "TIGHTER"

CLASS AND GRADE CORRESPOND

UNITS (ten thousandths)

6 tenths smaller than the nominal size, and therefore has the desired clearance.

INTERFERENCE FITS: In a similar manner, an interference fit would be written 123/ in which, by usage, the first number is the Class 1 hole and 23 the shaft. In this case, the shaft is to the left or plus side of the nominal line, which means it is bigger than the hole because the class numbers controlling its size are numerically greater than the class number of the hole.

TRANSITION FIT OR SELECTIVE ASSEMBLY: By the same reasoning, a selective assembly fit is written 110/ from which it is immediately apparent that the hole and the shaft have the same tolerances and must therefore be mated. The fit of the hole described in the above manner should be incorporated in the assembly or subassembly drawing, and the actual tolerances stated on the component drawing.

Ideal 7—Ball and Roller Race Housing Tolerances: Ball and roller race housings require three types of holes: (1) Oversize; (2) transition or bilateral; and (3) undersize holes. This system, being essentially a unilateral system, would not be complete if it could not provide suitable tolerances for ball race housings. Before proceeding, it is necessary to define the types of holes required:

1. *Oversize.* The maximum permissible diameter and the minimum diameter of the hole are both larger than the nominal size of the ball race to be used in it.
2. *Bilateral.* The maximum size is greater and the minimum size less than the nominal diameter of the ball race.
3. *Undersize.* Both diameters are smaller than the nominal outside diameter of the ball race.

With an oversize hole the smallest diameter of the hole is larger than the basic diameter, and therefore zero or 0 will not describe the minimum diameter. A number, which is the class number, is used to say how much the smallest diameter is larger than the basic diameter and therefore an oversize hole requires two numbers to define it.

In this system, since two numbers are used to describe a shaft, a prefix B is used to designate a bearing hole. An example will make this clear: It is required to describe a bearing hole for a 1½ inch outside diameter ball bearing with an easy sliding fit of + 24 tenths top limit, and + 12 tenths bottom limit. From memory, this will be a Class 8 and Class 4, or by calculation, 24 divided by 3 = 8, 12 divided by 3 = 4. Because both are plus and therefore to the left of

AMERICAN					
ABOVE	TO AND INCLUD'G	PILOT + 2 CLASSES FOR			
		XX	X	Y	Z
.029	.125	2	4	7	10
.125	.825	1	2	3	5
.825	.999	1	3	4	6
.999	1.510	1	2	3	4
1.510	1.999	1	3	4	5
1.999	2.510	1	2	3	4
2.510	2.999	1	2	4	5
2.999	3.999	1	2	3	4

BSI "U"		
FROM	TO	PILOT + 2 CLASS
0	.125	6
.126	.290	3
.300	.590	4
.600	.999	5
1.000	1.490	4
1.500	1.999	5
2.000	3.999	4

Fig. 5 — Replacement tables for American and British gage classes. Extra change steps are introduced owing to differing class intervals in the Pilot system

the oblique stroke, the bearing hole would be described accurately by B84/. In a similar manner a bilateral hole could be described by B2/1, and an undersize hole by B/14.

One of the salient features of the system is the mental picture conveyed by the position of the class numbers relative to the oblique stroke or basic line, indicating the category of the hole. Also, the magnitude of these numbers immediately shows the type of fit between the bearing and its housing.

Ideal 8—Manufacturing Tolerances for Gages: The usual practice of allowing the gagemaker 10 per cent of the limit as a manufacturing tolerance in a positive direction on the *go* end is adopted. Also, the 10 per cent manufacturing tolerance should be split on the *not go* end as shown in the following example: The gage to be made is nominally 1 inch in diameter with a 0.001-inch, or in other words, 10 tenths limit. Since 10 per cent of the tolerance can be allowed in the manufacture of the *go* end and, as already mentioned, this is used in a positive direction, the *go* end can be to a minimum size of exactly 1 inch in diameter, and a maximum size of 1 inch plus one tenth.

Here it must be stated that it is always the endeavour of the gage manufacturer to make the gage to the 1 inch plus one tenth, or top-limit diameter, with the objective of giving the maximum wear allowance or life.

Turning to the *not go* end, the manufacturing tolerance is split, that is to say, the *not go* end may be 5 per cent of the limit larger than its theoretical size of 1 inch plus one thousandth, or smaller than this size by 5 per cent of the limit.

Here again, the gage manufacturer endeavours to keep the gage up to the top limit, or its greatest permissible size, in this way ensuring that production has the greatest possible tolerance.

No mention has been made of "Workshop" and "Inspection" gage tolerances, for the reason that they are considered to be correct in theory, but impossible in practice. A new workshop *go* end is very slightly larger than its inspection counterpart but, owing to the more severe conditions under which the shop gage is used, wear is more rapid with the result that it is smaller than the inspection gage in a very short time, thus completely reversing this consideration. It may be argued theoretically that frequent inspection of the gages will prevent this, but in practice it must be admitted that such an ideal state is seldom, if ever, attained. With this in mind, and the fact that sharing the tolerance between inspection and shop robs the operator of much needed assistance, the Pilot + 2 system employs only one manufacturing tolerance.

In the chart, *Fig.* 2, of the complete system it will also be noted that the maximum manufacturing tolerance has been confined to two tenths, and this will be referred to later.

Ideal 9—Go Gage Wear Below Nominal: Established practice that the *go* end, the wearable member, should be allowed to wear a maximum of 10 per cent below the basic size before withdrawal from service is employed. To illustrate conditions in use, an example

with comments would serve a useful purpose.

A 1 inch, Class 1 gage has a limit of 3 tenths, and 10 per cent of 3 tenths is 3 hundredths of a thousandth, or 30 millionths. Under these circumstances the *go* end would be allowed to wear 30 millionths undersize, or below the basic diameter, before withdrawal from use. Since a certain amount of clearance under production conditions between the gage and the hole is required, it might justly be claimed that the hole will be, to all intents and purposes, size.

In the case of a 1 inch diameter gage with 0.001-inch or 10 tenths limit, the *go* end would be allowed to wear one-tenth below the basic diameter of exactly 1 inch. Space will not allow further explanations; suffice to say that this method is widely adopted in the engineering industry.

Ideal 10—American Limits: If this ideal can be fulfilled, then two important requirements are met: (1) Interchangeability is unaffected; and (2) American gages are not scrapped.

In American gage trade catalogs, particulars are given of the manufacturing tolerances for gages made to different grades of accuracy designated XX, X, Y, and Z. Since the practice of allowing 10 per cent of the limit in a plus direction on the *go* end as a manufacturing tolerance and ± 5 per cent of the limit on the *not go* end is adopted, a tolerance table can be constructed, *Fig*. 3.

Below the American tolerance table is placed the Pilot + 2 system from which it will be noted that there is a remarkable resemblance in the tolerance values employed. Further inquiry indicates, however, that the diameter steps at which the tolerance changes take place in the two systems are at variance, but this does not present a problem of practical magnitude. The similarities of the two systems as they now exist are so definite that the following is of interest:

XX Grade	Exact equivalents	71%
	One tenth difference	28%
X Grade	Exact equivalents	75.3%
	One tenth difference	12.3%
	Two tenths difference	12.4%
Y Grade	Exact equivalents	63.2%
	One tenth difference	36.8%
Z Grade	Exact equivalents	87.6%
	One tenth difference	12.4%

The percentages given are for holes 4 inches in diameter and less which, by investigation, comprise at least 98.4 per cent of all limited holes.

Exact equivalents for American tolerances are found in the new system to a very high degree as will be seen in *Fig*. 4. In the few cases where differences do arise, their magnitude is small when compared with the total tolerances employed and of no practical significance, the exception being three instances in the XX Class.

The replacement tables, *Fig*. 5, detail equivalents for American and BSI "U" tolerances from which it will be rightly assumed that the Pilot + 2 system sup-

plies the very important and extremely desirable connection between American and British practice.

Ideal 11—Gage Requirements: Reference has already been made to an upper gage manufacturing limit of two tenths on the *go* end, and to conserve space, the user's point of view only will be considered. A gage with a 5 thousandths limit under the 10 per cent rule could have a maximum of 5 tenths, or ½-thousandth wear allowance—in other words it could be this amount larger than the nominal size. Again, under the 10 per cent rule applying to the amount the gage can be allowed to wear below the basic size, another half-thousandth is available for increasing the life of the gage, or a total of 1 thousandth which by all standards of reasonableness is too great.

It will therefore be seen that if the maximum wear allowance above the basic size is confined to 2 tenths, then such a *go* gage end will be suitable for any class of hole whose limit is 2, 3, 4 thousandths or larger. This limitation reduces the cost of the gaging system, for it will be apparent that in the ideal situation when the *go* and *not go* gages are separate, as in larger sizes, one *go* end can be used for any hole with a tolerance of 2 thousandths and above. A series of *not go* gages only will be required to control the maximum size of the hole.

Ideal 12—Color Coding of Classes: Reference to *Fig.* 1 indicates that red is used for Class 1 gages, green Class 2, blue Class 3, amber Class 4, black Class 5 and above. This is carried into effect by using colored anodized aluminum handles which ensure that the gages are recognized instantly

Ideal 13—Reduction of Scrap: It is well known that a new, keen reamer cuts big, and let it be assumed that the hole is ½ inch in diameter and has been cut 8-tenths oversize, or in other words, a Class 4 hole. Under the color code, it follows that by painting the reamer amber between the top of the flutes and the shank, or by slipping on a colored synthetic rubber band, the reamer is immediately labelled as one cutting an 8-tenths oversize hole. As the reamer wears and the hole becomes progressively smaller, the color is changed at the various tolerance grades until red finally appears making it the most valuable reamer in the shop. In this way reamer life will be increased and scrap avoided.

Summary: Design of this new simplified system of limits was only finalized after an extensive and illuminating analysis of actual tolerances used throughout a wide cross section of industry. Thus, based on real engineering practice rather than theoretical suppositions, it solves in an eminently practical manner a knotty problem which has been with us too long. It is hoped that its utter simplicity and completeness will lend an attraction that, along with its high compatibility, will foster its widespread adoption.

Stamped Tape Replaces Painted Letters

USE OF stamped-in colored plastics film tape to provide recessed lettering in aluminum handles has increased production by 39 per cent and reduced labor costs by 33 per cent. Previously, the letters had been stamped and then spray painted.

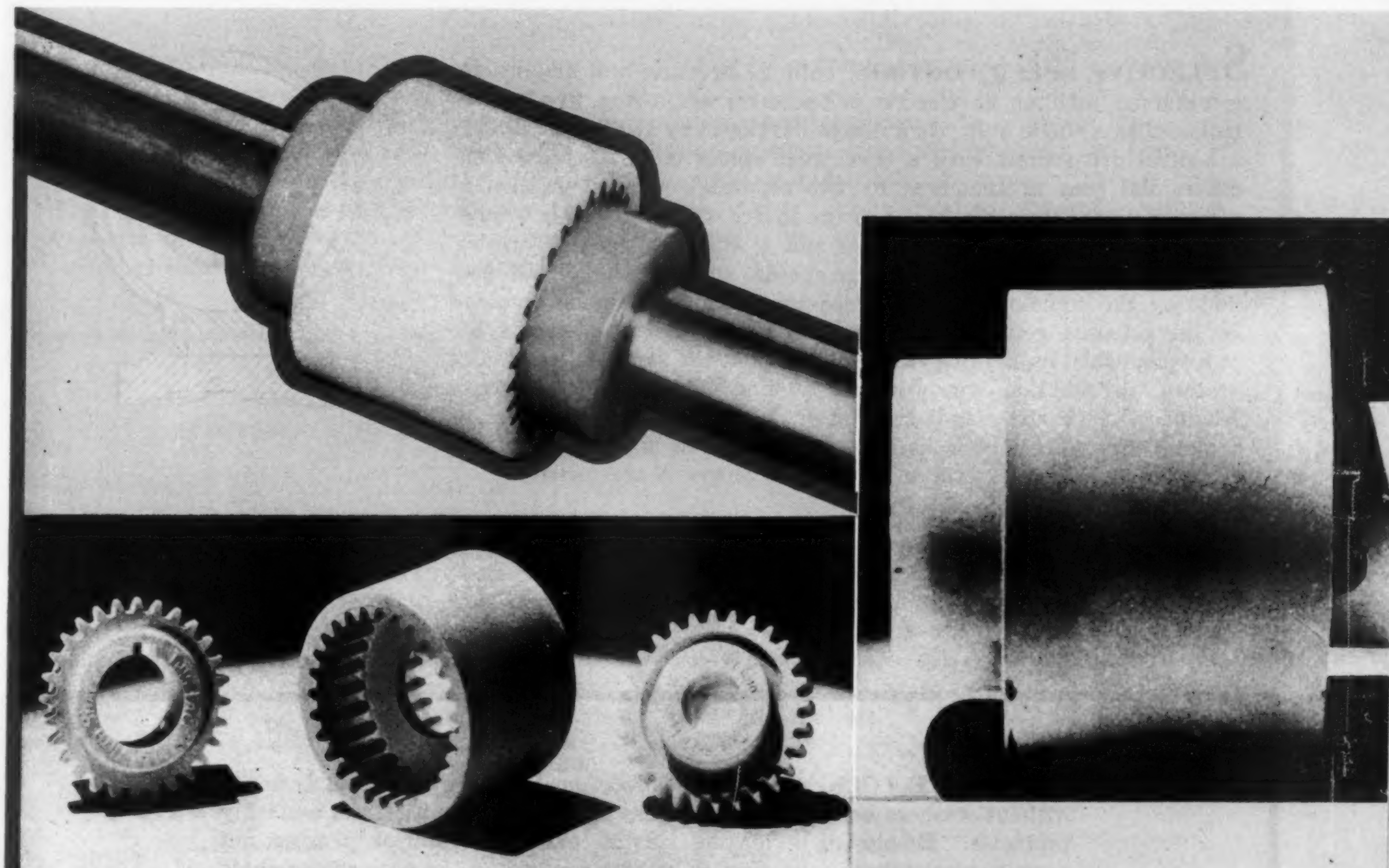
Handle output was 18 per man hour, and three employees were required for the earlier process, which included: (1) Application of a strip of paper masking tape, (2) Stamping of an impression through the tape, (3) Removing the masking tape from the recessed letters, (4) Spray painting the recessed letters, with the tape remaining on the handle acting as a mask, and (5) Removing the masking tape.

Two men now produce 25 handles per man hour using a three-step process:

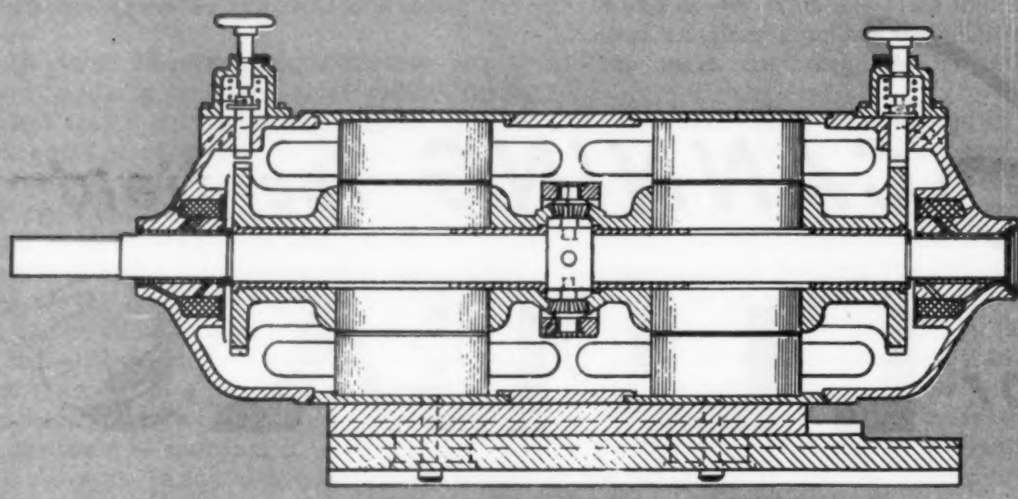
1. Applying a 1½ by 9-inch strip of yellow 3M plastic film tape.
2. Stamping the letters through the plastic tape.
3. Stripping off the tape remaining on the surface of the handle.

Tape stamped into the handle to a depth of approximately 1/16-inch remains in the handle, giving color to the letters. Additionally, the tape is resistant to acids, alkalis, water, salt water, alcohol, and hydrocarbons such as gasoline or kerosene.

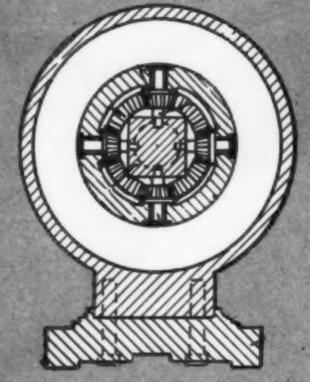
SCANNING the field for IDEAS



MINIATURE GEAR COUPLINGS, molded of nylon, simplify maintenance and assembly problems and reduce noise in fractional horsepower drives. Developed by John Waldron Corp. for bores in the ⅜ to ¾-inch range, a three-piece assembly consisting of two molded hubs and a molded connecting sleeve provides a flexible shaft connection which can be operated under misalignment. Torque capacity of the largest coupling, ¾ inch hub bore, is over 1100 pound-inches. Set-screw and keyway mounted, the coupling is light in weight and requires no lubrication. In addition to the cost reductions afforded through molding, the resilient nylon construction damps shock and vibration, resists abrasion and corrosion, acts as an electrical insulator and will withstand temperatures up to 250 F.

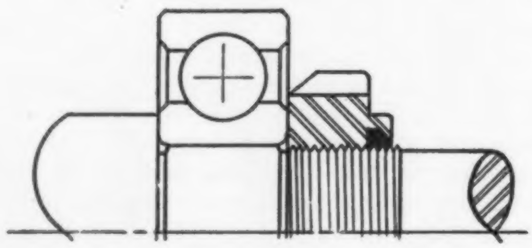


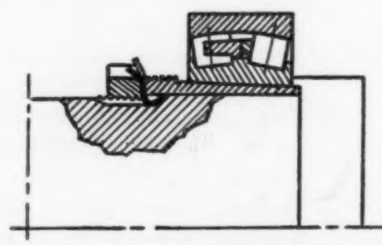
SELECTIVE SPEED CONTROL, both in direction and magnitude, is obtained with an ac electric motor drive employing two motors mounted as a single unit. In a design developed by William E. Raney, a central drive shaft with a bevel gear spider is driven through a differential gear arrangement by two separately excited multispeed rotors mounted to rotate freely on the shaft. Output speed is a function of the combined rotor speeds and is equal to one-half the algebraic sum of the individual rotor speeds; the variations made possible by the combination far exceed those which could be obtained by the separate rotors. Speed and directional control of each rotor is accomplished by varying the connections at the separate multipolar stators. In addition, spring-loaded latching devices permit selective locking of each rotor against rotation. Offering the advantages of a direct drive, the unit is light in weight and no larger in diameter than a conventional motor of equivalent power characteristics.

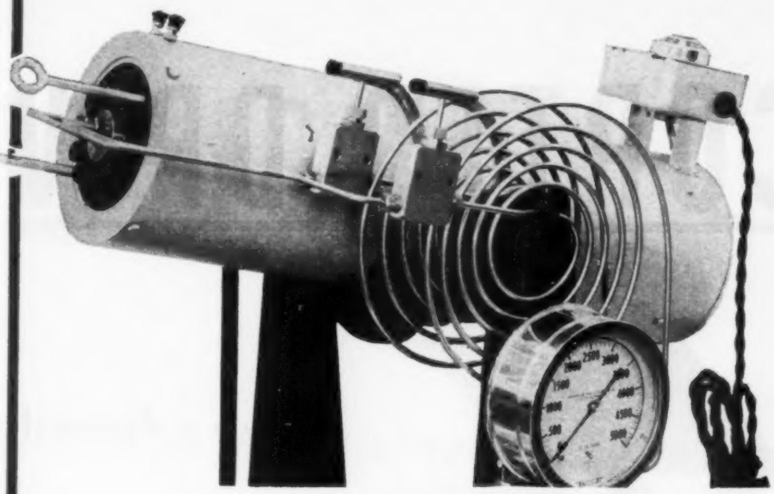


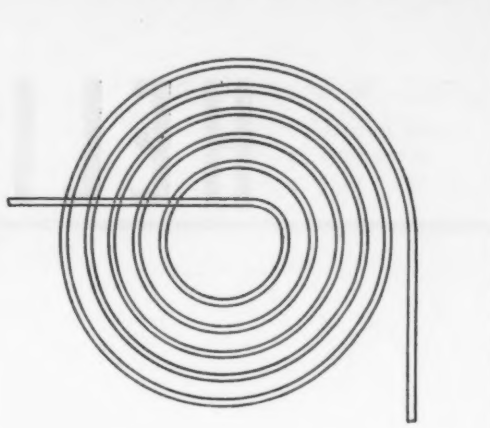
Pat. 2,578,837

ACCURATE LOCKING for preload or position of antifriction bearings without washers or keyways eliminates shaft discontinuities and assembly problems. Employing a locking elastic collar, the Shurlok bearing nut permits close-tolerance setting of bearing preload and service adjustments for wear takeup. Removal and reinstallation of the nut can be made up to ten times without loss of adequate locking torque.



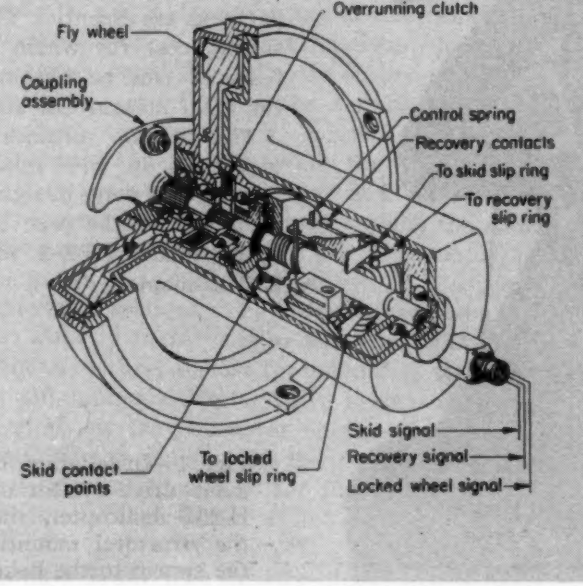


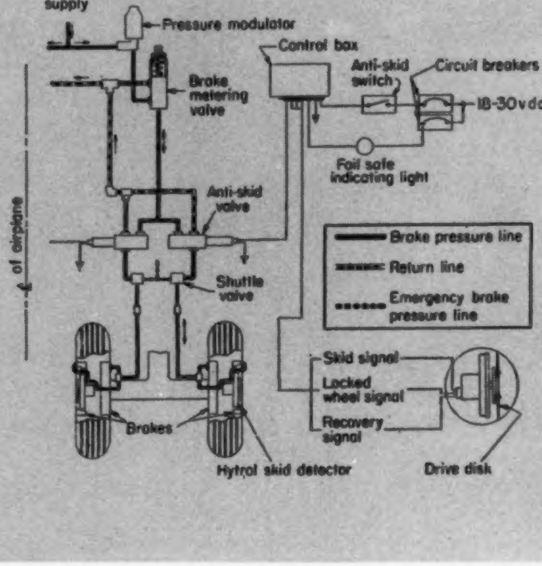




HIGH-PRESSURE CONTROL of gas flow to and from rocking chemical vessels operating at 5,000 to 60,000 psi is accomplished with a novel spiral tubing design developed by the Superpressure Div. of American Instrument Co. Employing ¼-in. OD stainless steel tubing, the design permits pressurizing of chemical reaction agitator chambers seesawing through an arc of about 30 degrees at rates of 26 to 58 times per minute. In addition, the flexibility achieved allows the reaction vessel to be swung into a vertical position, head up or down, for filling or emptying operations. Eliminating the packings and glands usually required, the design assures gastight sealing and facilitates assembly and maintenance.

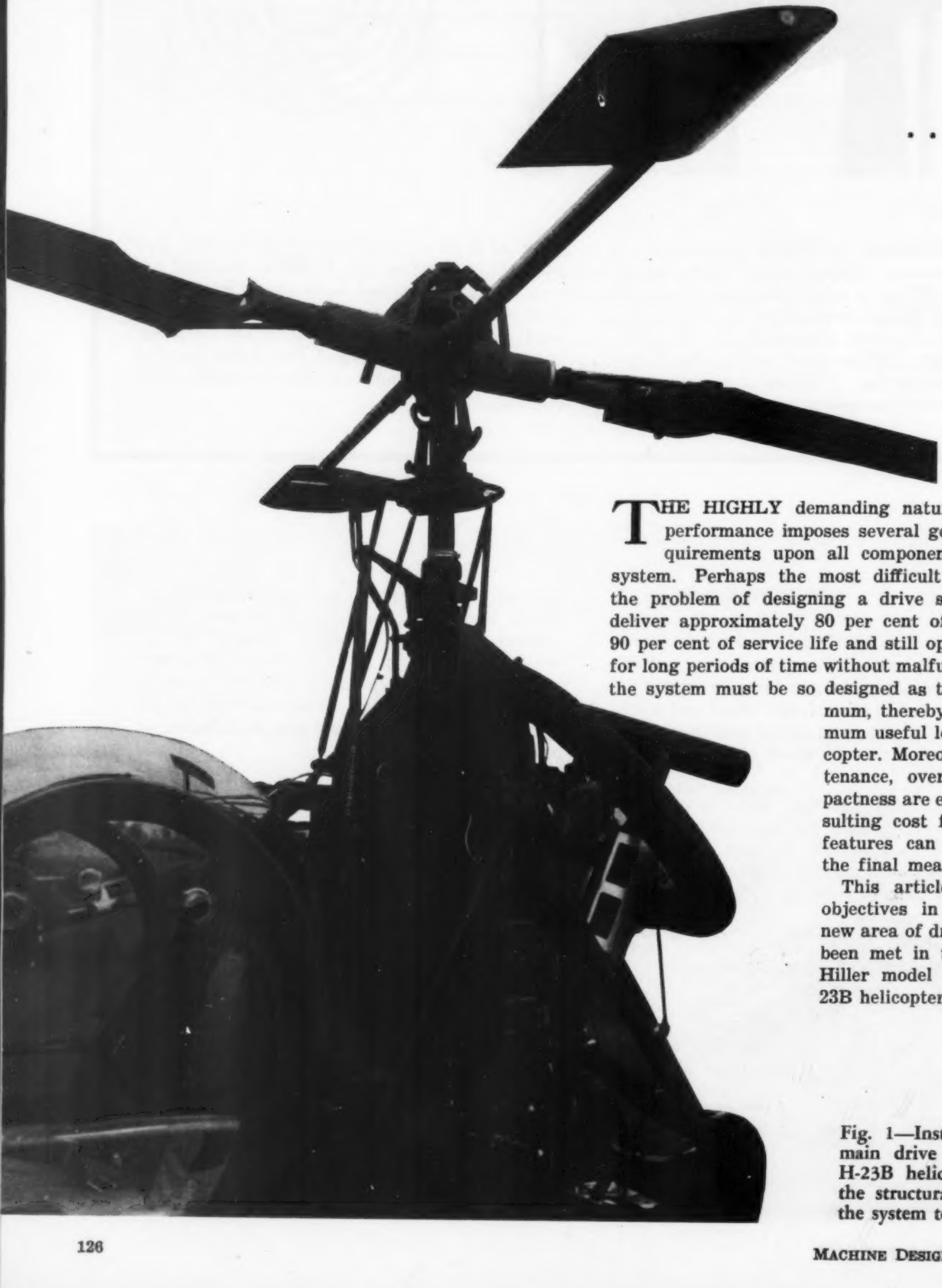
INSTANTANEOUS SPEED SENSING in an automatic control system for hydraulic aircraft brakes eliminates skidding and reduces maintenance and tire wear. The Hytrol system developed by Hydro-Aire Inc. employs an inertia flywheel, mounted in the aircraft wheel and driven through a slip clutch, to detect rotational speed as well as the rate of speed change. Signals transmitted to a control box and solenoid valve relieve and reapply metered brake pressure to prevent skidding and wheel locking. On-off braking at rates up to three times per second are achieved. Incorporating a fail-safe action, the system is employed as an auxiliary to the regular braking system and can be switched on or off at any time.





HELICOPTER DRIVE

. . . *demonstrates*



THE HIGHLY demanding nature of helicopter performance imposes several general design requirements upon all components of the drive system. Perhaps the most difficult requirement is the problem of designing a drive system that will deliver approximately 80 per cent of full power for 90 per cent of service life and still operate efficiently for long periods of time without malfunctioning. Next, the system must be so designed as to weigh a minimum, thereby insuring maximum useful load for the helicopter. Moreover, easy maintenance, overhaul and compactness are essential. The resulting cost for which these features can be obtained is the final measure of success.

This article outlines how objectives in this relatively new area of drive design have been met in the case of the Hiller model HTE-2 and H-23B helicopter, *Fig*. 1.

Fig. 1—Installation of the main drive system on the H-23B helicopter, showing the structural mounting of the system to the helicopter

E DESIGN

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ite

application of diverse components for unusual performance conditions

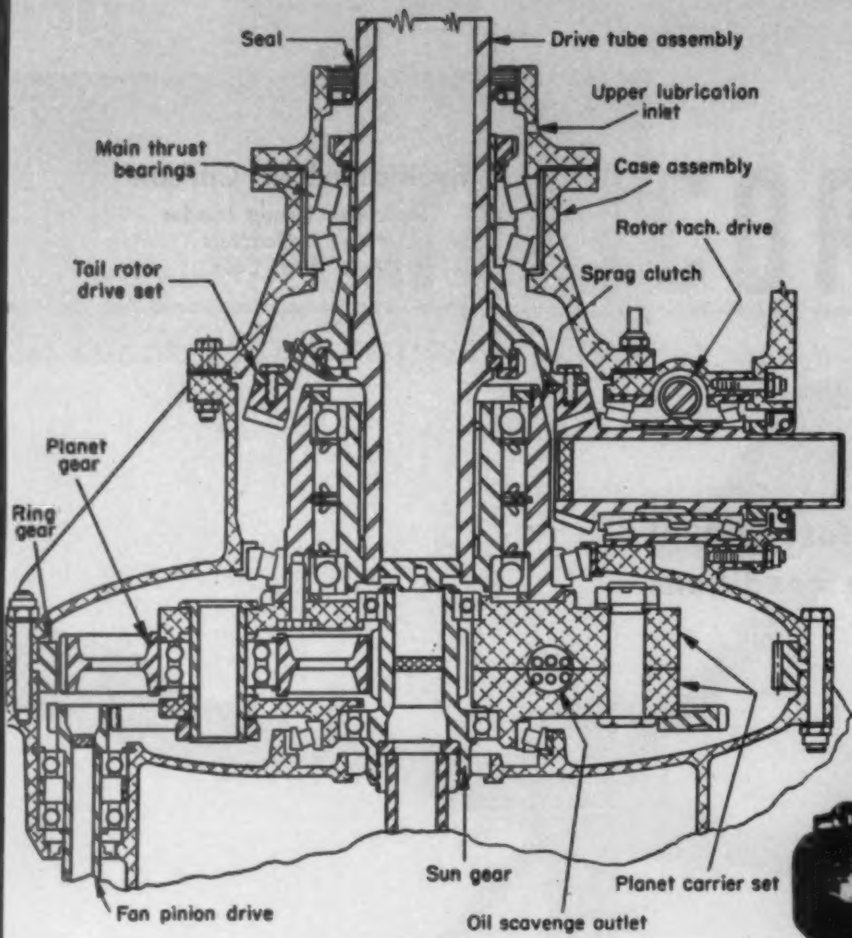
Principal design features might best be first described. Power is received from the engine at high speed and low torque and is converted, by the transmission, to low speed and high torque. The low speed value is dictated by aerodynamic requirements of the main rotor. The power is then delivered directly to the main rotor and through additional gear trains to the tail rotor, cooling fan, generator, and tachometer drive.

Two separate engaging devices are provided in the main rotor drive system: an automatic centrifugal shoe clutch, which satisfies the rotor starting requirements, and a free-wheeling clutch, located on the low-speed side of the transmission, which automatically disengages the engine to ensure rotor autorotation in the event of power failure. The main transmission, or power train, consists of a single-stage planetary spur system, while the tail rotor and accessory drives are of both spur and bevel single-mesh arrangements. The planetary system, the accessory drives and the two clutches are all mounted and enclosed in the transmission case, which consists of three individual sections. The lower transmission case is bolted directly to the engine mounting plate, providing the structural means for mounting the entire power plant unit, main rotor drive system and main rotor system on the helicopter engine mounts. The center and upper sections of the transmission case serve primarily as mountings for the various components of the transmission.

Main Power Train: The fundamental design requirement of the main power train was provision of a means of transmitting the 200 hp from the Franklin model 0-335-6 engine to the main rotor at 338 rpm. Since the engine operates at 3100 rpm, a speed reduction of approximately 9.2/1 was required. The reduction of speed, in itself, posed no particular problem, for any number of gear systems could have provided it. Problems of weight, simplicity and compactness, however, dictated design.

A single-mesh gear train arrangement was considered and eventually eliminated, for while it possesses extreme simplicity it also becomes quite heavy and cumbersome. For compactness, recourse was made to the internal gear systems such as the differential train and the planetary train. Of the two systems, the differential train makes possible the greatest speed reduction in the minimum space, but because of the excessive tooth loads and high engagement velocities it becomes somewhat impractical for high power transmission. With the field narrowed to a planetary system, the compactness and weight advantages of the compound planetary were weighed against the simplicity and lower cost of the single-stage planetary. From this analysis, finally, the single-stage planetary system, *Figs.* 2 to 4, was selected.

As shown in *Fig.* 2, the sun gear is the driving gear and is supported in the planet carrier by two single-row ball bearings. These bearings also provide for relative rotational motion between the carrier and the sun gear. The planet carrier is the driven member supporting three bearing shafts on which the three planet gears are mounted. Each planet gear



HELICOPTER DRIVE DESIGN

Fig. 2—Above—Cross section of the main drive system showing principal components

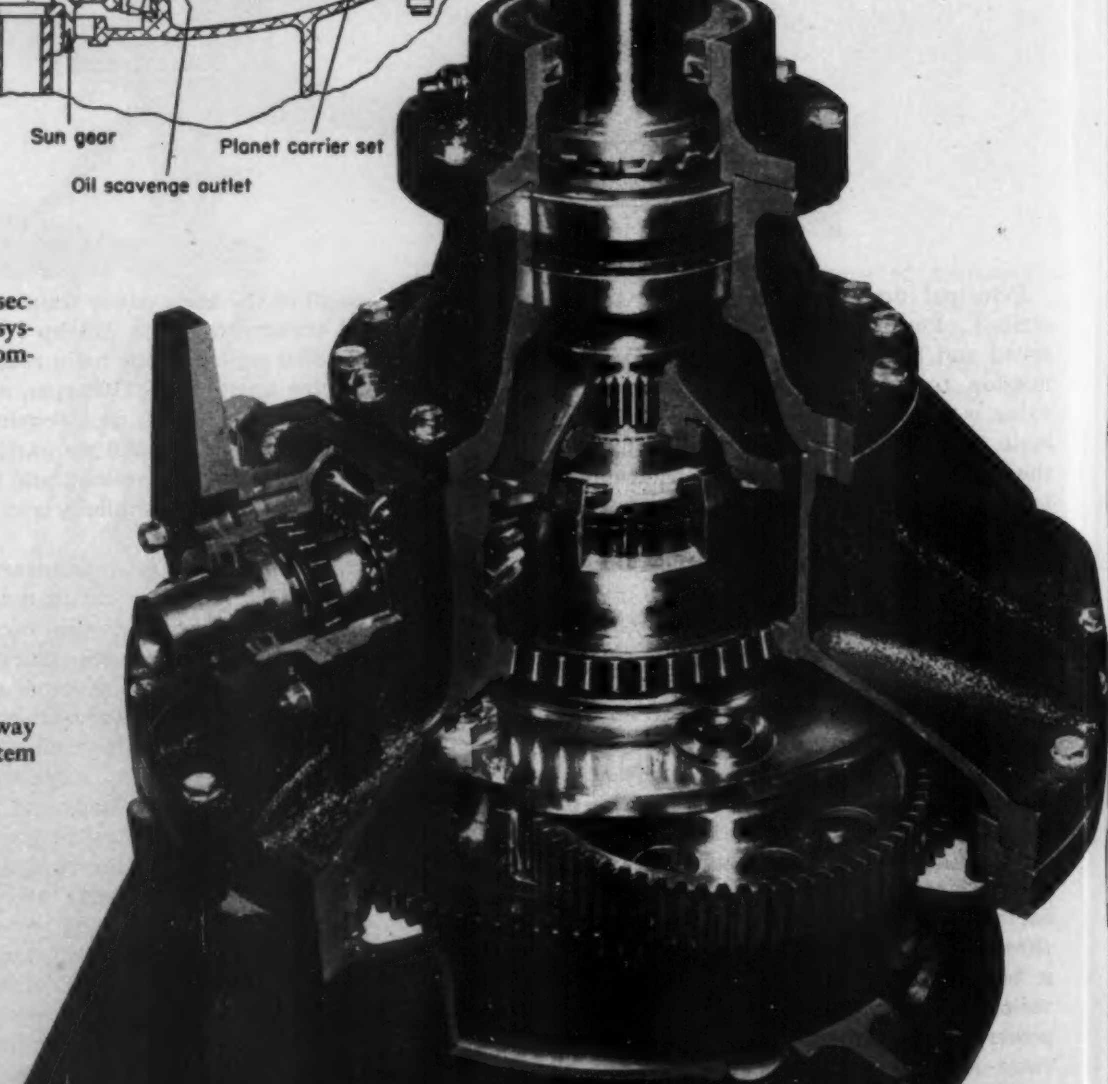


Fig. 3 — Right — Cutaway view of main drive system

Table 1—Main Rotor Drive System Gears

Design Property	Main Power Train Sun	Main Power Train Planet	Main Power Train Ring	Tail Rotor Power Gear	Tail Rotor Power Pinion	Rotor Tachometer Driving	Rotor Tachometer Driven	Fan Drive Gear	Fan Drive Pinion
Gear Type	20-deg full-depth spur			17½ deg Gleason generated straight bevel		14½-deg right-angle helical		20-deg full-depth spur	
Diametral Pitch	10			8		24		10	
Number of Teeth	18	65	147	58	19	27	18	126	13
Material	SAE 3317	SAE 3317	Nitralloy G	SAE 3317	SAE 3310	SAE 4140	SAE 4140	SAE 4620	SAE 4620
Core Strength (1000 psi)	165–195	165–195	125–145	150–190	150–190	125–140	90–150	150–190	150–190
Case Hardness (Rockwell C)	60–63	60–63	65–67	60–64	60–63	50–55	50–55	60–63	60–63
Case Depth (inches)	.035–.045	.035–.045	.015–.020	.030–.035	.030–.035	....	....	.025–.030	.025–.030
Engagement Velocity (fps)	1300			640		630		1115	
Face Load (lb per in.)	1270	1510		1000		....		590	

rotates on a double-row radial ball bearing. In mesh with the three planet gears, and mounted between the center and lower transmission case, is the internal or ring gear which serves as the stationary member of the train.

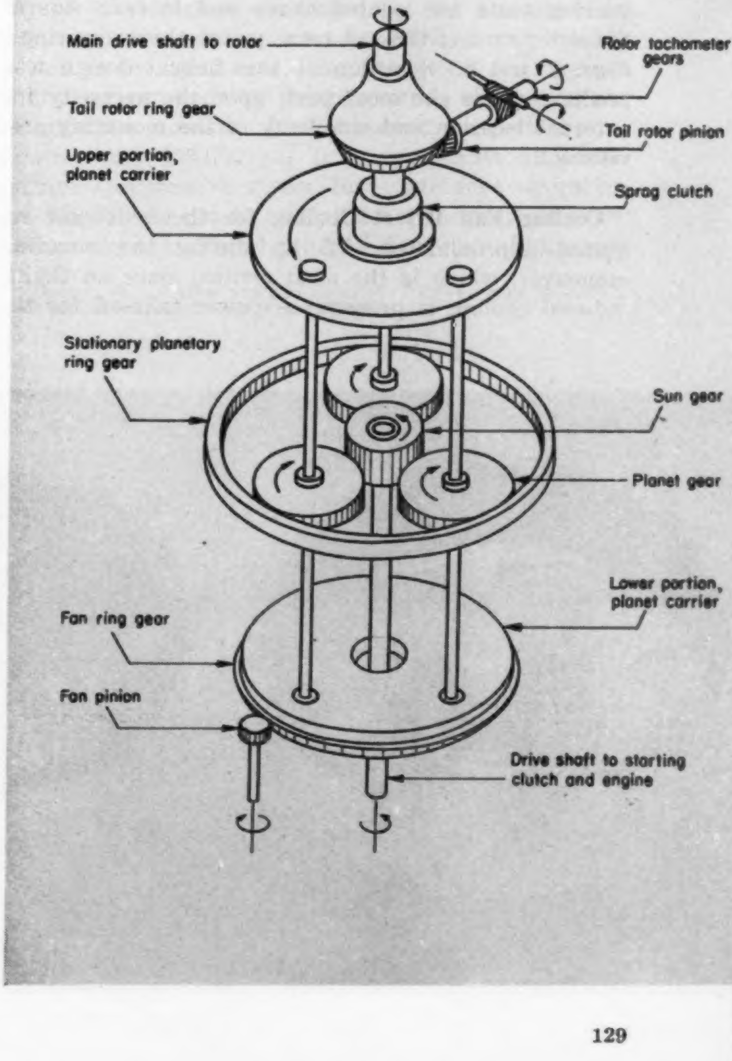
Conventional design methods were employed throughout the entire system, and service experience, to date, has substantiated their adequacy. The strength of the power train was determined through utilization of the standard Lewis stress and Buckingham dynamic load relationships. Case depth and hardness of the gear teeth were selected on the basis of the surface and subsurface stresses, which were predicted with the well-known Hertz equations. TABLE 1 lists the pertinent characteristics of the main power train components.

Tail Rotor Power: In addition to performing its prime function, that of delivering power to the main rotor, the drive system must also deliver power to the tail rotor for antitorque and directional control purposes. The tail rotor power take-off must be located on the main rotor side of the free-wheeling clutch since directional control is required during autorotation as well as powered flight. The tail rotor installation required approximately 10 hp during 80 per cent of helicopter operation and 18 hp for the remaining time. Aerodynamic considerations dictated a tail rotor speed of 1950 rpm or a ratio of 5.76/1.0 between the tail rotor and the main rotor. However, the main transmission power take-off provides only a portion of this speed change, the remainder being effected in the tail rotor gear box.

In these helicopter models, the tail rotor power is obtained through a set of straight bevel gears, ***Figs.*** 2 and 5. The gear is located adjacent to and above the free-wheeling clutch, and is mounted on the face of a steel hub which is splined to the main rotor drive shaft. The upper portion of this hub serves as the inner race of the main helicopter thrust bearings. The pinion axis is at 90 degrees to the gear axis and is supported by two thrust type roller bearings which are mounted in the center transmission case.

For this particular application smoothness and quietness were not prime objectives, and the speed reduction was small. Therefore, straight bevel gears were selected, in lieu of Zerol bevels, spiral bevels or hypoids, primarily because of the simplicity of the straight bevel set and its resulting economy. In general, straight bevel gears are utilized in applications where the peripheral speed in feet per minute and rpm are below 1000. The tail rotor power take-off pinion has a peripheral speed of 640 fpm, TABLE 1, and operates at 1033 rpm. The use of straight bevel gears in this application has so far proved quite satisfactory.

Fig. 4—Schematic of gearing arrangement



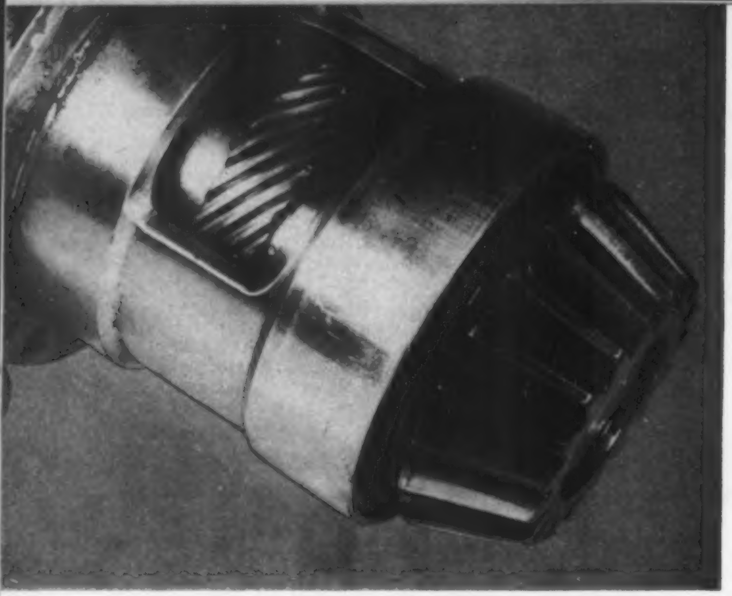


Fig. 5 — Tail rotor pinion and tachometer drive gear

Splined to the tail-rotor pinion and serving as a spacer between the pinion thrust bearings is the helical driver gear for the rotor tachometer. The speed is increased through this gear set by a factor of 1.5, permitting the rotor tachometer to operate at one-half engine speed, 1550 rpm. This speed is required since both rotor and engine speeds are indicated on the same instrument to provide for observation of engine and rotor synchronization. The helical follower, whose axis is at 90 degrees to that of the driver, is mounted in two radial ball bearings. The bearing seats are located above and to each side of the outer race of the tail rotor pinion thrust bearings, *Figs.* 2 and 5. Selection of this helical design was predicated, for the most part, upon the necessity for external location and simplicity of the mounting provisions.

Cooling Fan Drive: Cooling for the helicopter required approximately 7.5 hp during the hovering maneuver, which is the most critical since no flight-induced cooling is present. A power take-off for the cooling fan, which supplies engine cooling and oil cooling air, was located on the low-speed side of the main power train for simplicity and availability for maintenance. The driving member of the power take-off is a large spur gear bolted directly to the main planet carrier. The pinion, a very small spur gear, is mounted in two radial ball bearings which are seated directly in the center section of the transmission case. With this arrangement, a speed increase to the fan of 9.7/1 is effected.

Simplicity and economy were both attained through the use of the conventional spur gear design; however, the system suffered a severe weight penalty as a result of an indeterminate and somewhat inadvertent transmission of power during certain transient drive conditions. A strength margin of safety of approximately 2.00 was designed into the entire fan drive system.

Starting Clutch: Helicopters, which are powered by reciprocating engines, require starting clutches for two basic reasons. First, they separate the engine from the drive system at starting engine speed, thus permitting engine warm-up without turning of the rotors.

Second, starting clutches protect transmission and rotor from irregular starting loads. Selection of a particular type of clutch for this application was made after consideration of the various design requirements. Some of these requirements are as follows:

1. The clutch should engage at a certain predetermined engine speed and operate freely at all lower speeds.
2. The clutch should, if possible, engage automatically.
3. The clutch should be of the friction type, allowing some slip while the rotor is being brought up to speed.
4. Automatic compensation for wear should be provided.
5. The clutch should be as light as possible.

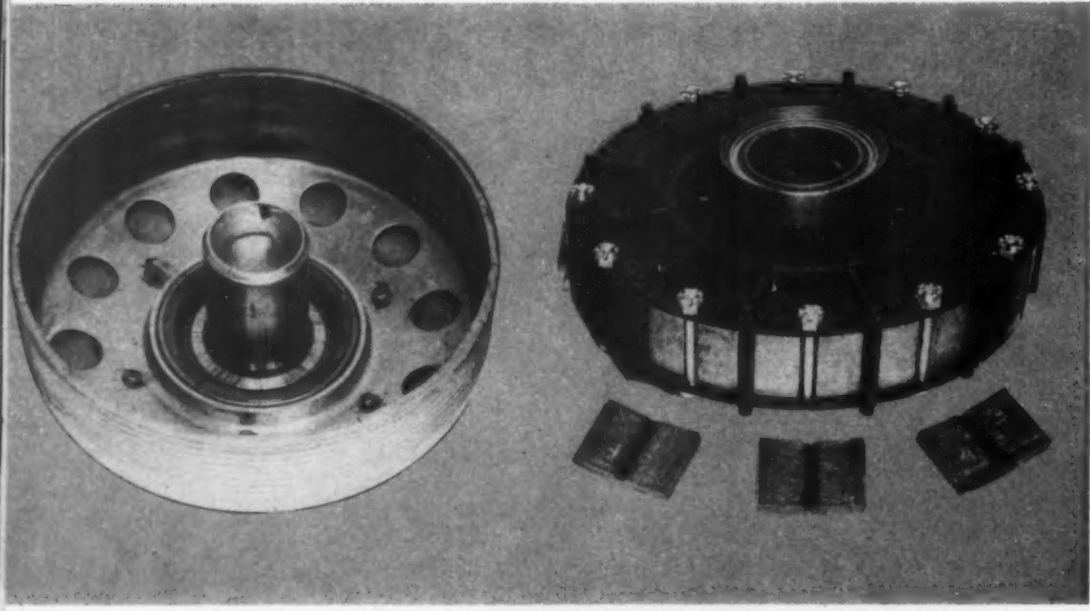


Fig. 6—Automatic centrifugal (mercury) starting clutch, shown partially disassembled

The automatic centrifugal, or mercury, shoe clutch, *Fig.* 6, met the above requirements in the most satisfactory fashion and was selected for use on the H-23B and HTE-2 helicopters.

The lower portion of the clutch, which is integral with the engine, is bolted directly on the upper face of the flywheel. The isolated upper portion of the clutch rotates on two radial ball bearings and is splined directly to a short intermediate shaft. The bearings are seated in the lower portion of the clutch. The intermediate shaft is in turn splined to the sun gear of the main planetary system. This small intermediate shaft provides for any angular and radial misalignment between engine and transmission. As its name implies, the clutch operates by the action of 12 individual spring loaded shoes which are actuated by the centrifugal pressure of the mercury. The springs are simply preloaded so that when the engine attains 600 rpm the individual shoes proceed to move radially outward and smoothly engage with the upper portion of the clutch. Engagement (no slippage) is completed at 1600 rpm.

Since the clutch is on the high-speed side of the main power train, it must be capable of transmitting 200 hp at 3100 rpm. The clutch selected is actually rated at 1250 ft-lb at 3100 rpm, about 3.5 times operating torque. Transmittal of this torque indicates a shoe pressure of 260 psi and a calculated friction coefficient of 0.35.

Free-Wheeling Clutch: For successful operation in the autorotation maneuver, helicopters must have a provision for automatically allowing the rotor to function as an autorotating windmill rather than as a propeller. In these helicopter models a sprag type overrunning clutch, *Fig.* 7, serves this purpose. The outer race of the clutch is attached directly to the main planet carrier while the inner race is splined to the main rotor drive shaft. Placing the clutch on the low-speed side of the main power train subjected it to the maximum torque conditions, but this penalty was accepted in order that the autorotation per-

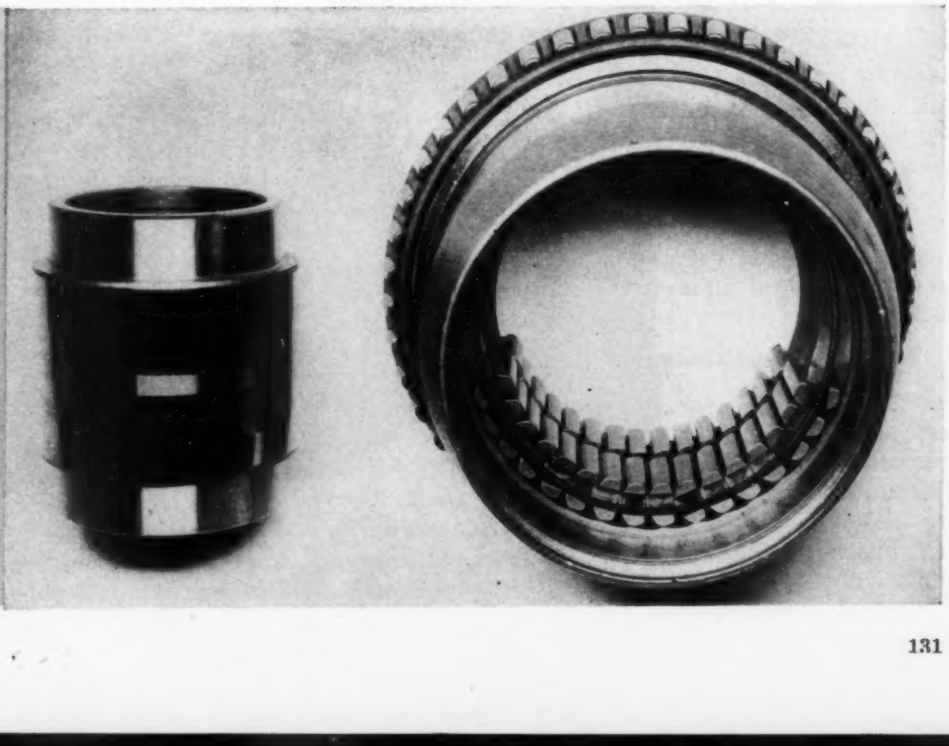
formance might be insured in case of transmission malfunction. Also, autorotation performance is improved since the main power train is not driven during the maneuver.

A number of overrunning clutch types were available for this application, such as cam roller, ratchet, friction and sprag clutches. A study of the different types indicated that of those available the sprag and cam roller clutches presented distinct advantages from the standpoint of apparent torque capacity, reliability, weight and compactness. Further study of these two types revealed that the sprag clutch was inherently the superior of the two in terms of torque capacity. Detailed analytical comparison of the two types has already appeared in the literature.* *Fig.* 8 shows the operating principle of the sprag clutch. Rotation of the inner race moves the sprags into a position where they are wedged between the inner and outer race. The torque is then transferred by means of the friction components of the normal forces which exist on the sprag units. Grooves in the upper and lower faces of the sprag hold a garter spring which energizes all of the sprags as a system so they will rotate and pick up load simultaneously.

Situated on the high torque side of the transmission, the clutch is designed for an operating torque of 56,000 lb-in., which results in a surface compressive (Hertz) stress of approximately 450,000 psi. Capacity of the clutch is controlled by the magnitude of the subsurface shear stresses, which are inevitably critical in the inner race. Magnitude of these stresses is directly reflected by the magnitude of the surface compression stresses. Previous design experience indicates that 400,000 psi is a reasonable allowable surface compressive stress for continuous operation and 650,000 psi can be tolerated for intermittent overload conditions. The depth and hardness of the case

* Dietrich W. Botstiber and Leo Kingston—"Freewheeling Clutches," MACHINE DESIGN, April, 1952, Pages 189-194.

Fig. 7 — Double - row sprag overrunning clutch used in initial production



for the races and sprags are controlled according to the variation as well as the magnitude of the subsurface shear stresses. However, for most conventional designs, which are based on the suggested allowable surface stresses, a hardness of 60 to 63 Rockwell C and a case depth of 0.09 to 0.10 inch seem to be acceptable minimums where depth is defined as the region where hardness is above 50 Rockwell C. Strength of the races is not a problem, for their configuration is governed by rigidity considerations.

Design Changes: Service experience on the H-23B and HTE-2 helicopters, which now includes machines with over 300 hours of flight, represents an encouraging measure of design success. The changes which have been incorporated in the main rotor drive system as a result of this experience have, for the most part, been of a minor nature.

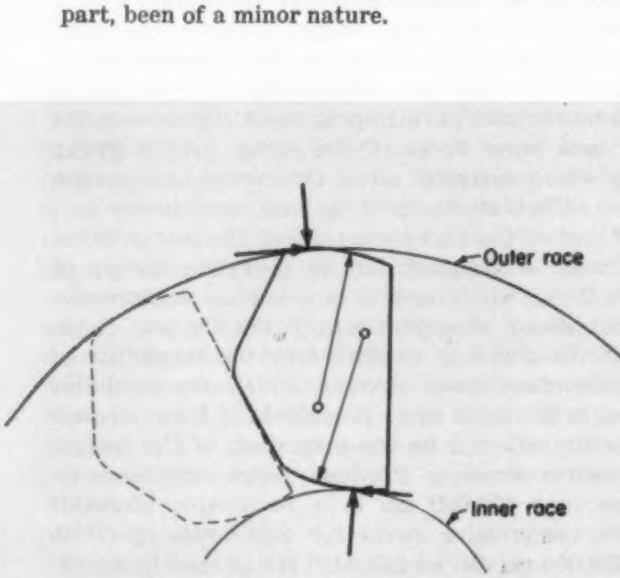


Fig. 8—Sprag clutch detail

Early in the operation of the helicopter two adjustments to gear tooth configuration were necessary to improve service life: (1) The tooth face width of the large driving spur gear of the cooling fan power take-off was increased from 0.375 inch to 0.500 inch; (2) the teeth on the main power train sun gear were crowned 0.0015 inch to prevent the occurrence of large concentrated loads on their outer edges. Both changes have resulted in a desirable improvement of service life.

In the original design, lubrication of the entire main rotor drive system was provided by a system which was integral with the engine system. The oil was introduced into the transmission at a point just above the main thrust bearings and scavenged at a point in the lower transmission case, *Fig.* 2. Service experience, supported by a series of test stand runs, indicated that the tail rotor power take-off and sprag clutch were not being sufficiently lubricated. As a result, excessive wear occurred on the tail rotor pinion and corrosion formed on the sprag units. Additional lubrication at this point was introduced by an oil jet, mounted in the center transmission case and adjacent to the tail rotor pinion. This change has resulted in satisfactory lubrication of the bevel gear set and the sprag clutch.

Perhaps the most significant information to emerge from the helicopter's operation pertains to the free wheeling sprag clutch. Under certain extreme, transient, torque loading the clutch would slip and momentarily release the torque load. Moreover, under similar conditions, the clutch would overturn; sprag units rotated over center, preventing free wheeling. A series of exploratory dynamic and static tests revealed that several sprag units were carrying the torque load instead of all sprags sharing the load. As a result of these findings, the clutch manufacturer developed a double-row caged sprag clutch which incorporates a system whereby all of the sprags are forced to carry their share of the applied torque, *Fig.* 9. Recent preliminary tests seem to indicate that the double-row caged sprag unit is superior to the original.

Fig. 9 — Experimental double - row caged sprag clutch designed to insure equal load distribution among all sprags

Fig. 1 — Landing gear strut composed of tubular members loaded primarily in bending

DESIGNING Cylinders and Struts —for maximum strength

How to realize optimum strength-weight ratio for cylinder tubes and struts through an analysis based on the practical limitations imposed by machinability requirements and permissible deformation

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IN THE DESIGN of aircraft parts, because minimum weight is of primary importance, design efforts must be directed always toward achieving the highest possible strength-weight ratio. However, an increase in strength does not always necessarily result in a weight reduction. For example, it is sometimes impossible to take advantage of the increased strength of a tube resulting from higher heat treating values because of the inability to finish machine the part.

In this article it will be shown that the cylinder tubes in hydraulic actuators, *Fig.* 1, designed for minimum weight have a definite practical limit of strength beyond which no advantage can be gained. Also, the general cases of tubes in bending and tubular columns will also be discussed.

Hydraulic Actuating Cylinders: Practical limits for the maximum strength of cylinder material can be established by consideration of hoop tension due to internal fluid pressure, machining limits and allowable diametral deflection. In general, the minimum diameter of the cylinder tube is fixed by function and then the minimum wall thickness and practical maximum strength are determined. These strength limits are especially useful because of the additional costs which can result, with no weight reduction, in machining harder material, heat treating for higher values, refining the design to avoid stress concentrations, fatigue, etc.

The extent to which a tube can be machined depends upon the material, diameter, length, manufacturing methods, available tools and the skill of the machinist. However, for simplicity, a maximum diameter to thickness ratio, *K,* is sometimes used as a measure of the machinability. This limit usually has additional significance from the standpoint of local effects.

Basic considerations in the design of a hydraulic

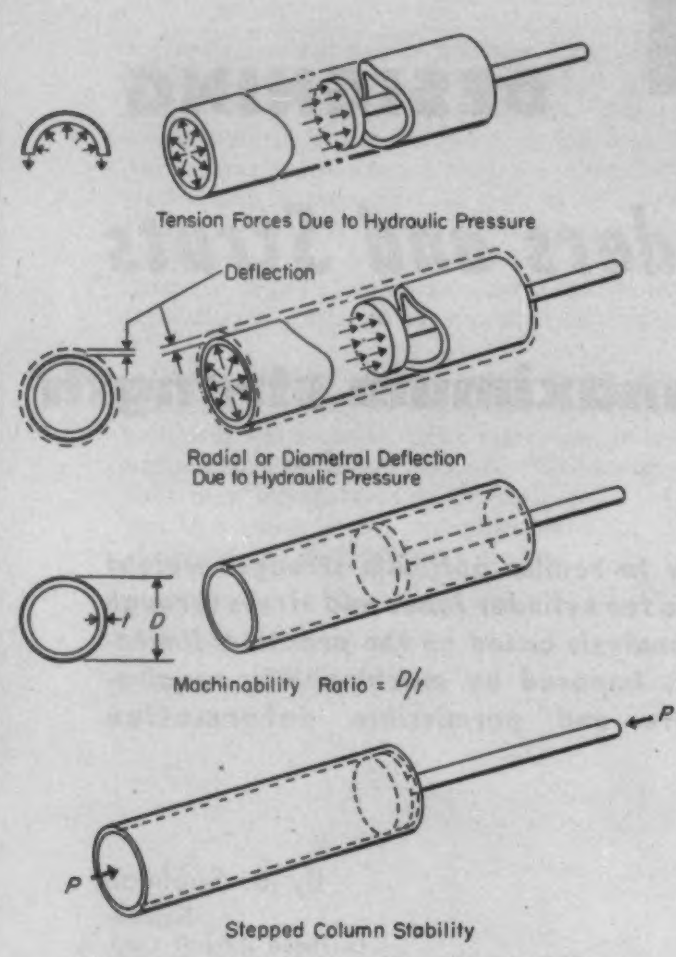


Fig. 2—Factors to be considered in the design of cylinder tubes of maximum strength

actuating cylinder tube, *Fig.* 2, are:

1. Hoop tension due to internal fluid pressure
2. Machinability limitations
3. Diametral deflection
4. Overall column stability.

A practical maximum strength value for cylinder tubes can be readily derived. As shown by Roark[1],

$$\sigma_u = \frac{p F_1}{2} \left(\frac{D}{t} \right) \quad \text{(1)}$$

where symbols are as defined in the Nomenclature. The maximum value of p, usually the relief valve setting, is approximately 1700 and 3100 psi for 1500 and 3000-psi hydraulic systems, respectively.

Since the maximum $D/t = K$, the minimum value of t for a given value of D is

$$t = \frac{D}{K} \quad \text{(2)}$$

There have been a few instances where, in the interest of weight reduction, values of K in the neighborhood of 60 have been achieved. However, in the case of steel, 50 is a commonly used limit.

Substituting the values of p and K in Equation 1, the practical maximum ultimate strength limits for cylinder tubes in 1500 and 3000 psi hydraulic systems are:

$$\sigma_u = \frac{1700\ (2.5)}{2}\ (50) = 106{,}000 \text{ psi}$$

$$\sigma_u = \frac{3100\ (2.5)}{2}\ (50) = 194{,}000 \text{ psi}$$

To meet functional requirements of hydraulic cylinders, *Fig.* 3, particularly in regard to the packing, it is necessary to limit the diametral deflection of the cylinder tube. However, there is no uniform agreement as to limiting values. MIL-P-5514 A[2] states that the diametral expansion of the cylinder shall not exceed 0.002-inches per inch of bore at the midpoint of the cylinder. The Lockheed Hydraulic Staff Office Design Manual recommends that the deflection of the inside diameter shall not be in excess of 0.0075-inch if leather backup rings are used, and 0.004-inch if they are not. These values are independent of diameter and occur at system pressure.

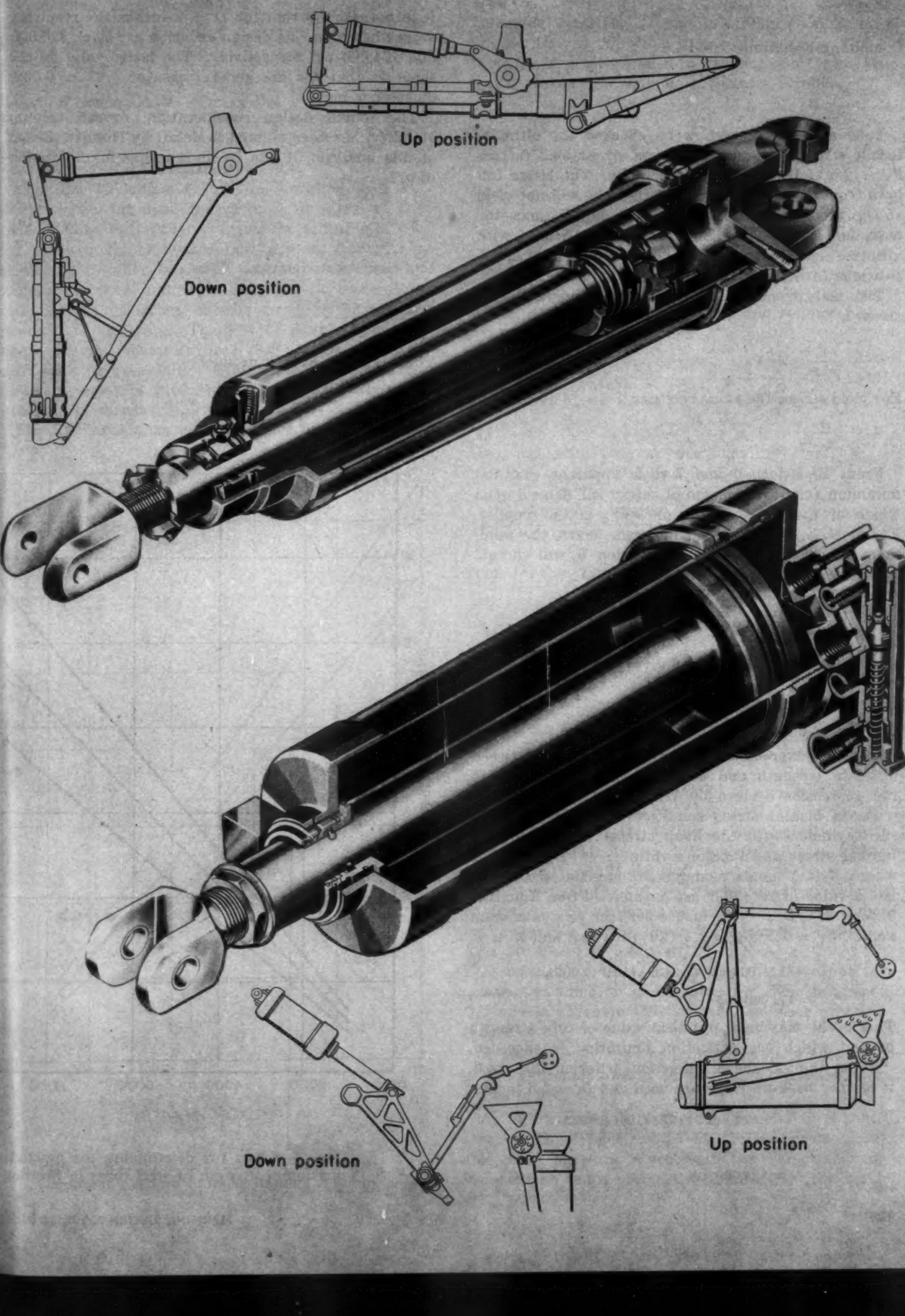
A general relationship between maximum practical strength and diametral deflection can be developed

Nomenclature

A = Cross sectional area, square inches
a = Effective length of cylinder, inches
B, C = Constants
D = Outside diameter of tube, inches
E = Young's modulus of elasticity, psi
E_h = Effective Young's modulus in hoop direction, psi
F_1 = Ultimate load factor
F_2 = Proof load factor
I = Moment of inertia, inches^4
$K = D/t$ = Machinability limit
L = Column length, inches
M = Bending moment, pound-inches
m = Subscript for effective stiffness factor, defined in Equation 12
N = Margin of safety
n = Factor defined by Equation 10
P = Applied load, pounds
P_{cr} = Critical buckling load, pounds
p = Internal fluid pressure, psi
t = Tube wall thickness, inches
δ = Diametral deflection, inches
ϵ = Unit diametral strain, inches per inch
σ = Applied column stress, psi
σ_{cr} = Critical buckling stress, psi
σ_t = Applied hoop stress, psi
σ_u = Ultimate allowable tensile stress, psi
σ_y = Tensile yield stress, psi
σ_b = Applied bending stress, psi
σ_{bn} = Bending modulus, psi

[1] References are tabulated at end of article.

Fig. 3—Typical aircraft applications of hydraulic cylinders. Diametral deflection of the cylinder tubes must be limited to assure effectiveness of packings



by a procedure similar to that used by Williams[3]:

$$\sigma_t = E_h \, \epsilon \qquad (3)$$

Also

$$\epsilon = \frac{\delta}{D} \qquad (4)$$

Combining Equations 3 and 4,

$$\sigma_t = E_h \frac{\delta}{D} \qquad (5)$$

Tube strength must now be checked for ultimate tensile stress, σ_u, and yield stress, σ_y, at load factors $F_1 = 2.5$ and $F_2 = 1.5$, respectively. The latter follows from the requirement that the tube will not yield at the proof load which is defined as 1.5 times the maximum applied load. The factor of 2.5 used with ultimate tensile stress is a requirement in most specifications for hydraulic cylinders.

The margin of safety based on ultimate tensile stress is

$$N = \frac{\sigma_u}{\sigma_t F_1} - 1 \qquad (6)$$

For yield stress, the value becomes

$$N = \frac{\sigma_y}{\sigma_t F_2} - 1 \qquad (7)$$

From Equations 6 and 7 it is apparent that the minimum (critical) margin of safety will depend upon which of the ratios, σ_u/F_1 or σ_y/F_2 is the smaller. For steel, σ_u/F_1 is the smaller and, hence, the minimum margin expressed by Equation 6 will dictate the design of the cylinder.

From Equation 6

$$\sigma_u = (1 + N)\, F_1\, \sigma_t \qquad (8)$$

Combining Equations 5 and 8 gives

$$\sigma_u = (1 + N)\, \frac{F_1 E_h \delta}{D} \qquad (9)$$

Equation 9 expresses the desired relationship between ultimate strength and deflection. It is identical to the expression derived by Williams[3].

For a biaxial stress condition in a circular pressure cylinder with the hoop stress twice the longitudinal stress and Poisson's ratio $= 1/3$, $E_h = 1.2\, E$ where E is Young's modulus for uniaxial stress and has a value of $29\,(10)^6$ psi for steel. From Equation 9, for $\epsilon = \delta/D = 0.002$-inches per inch as a maximum value[2], $F_1 = 2.5$, $E_h = 1.2\,(29)(10)^6$ psi and $N = 0$,

$$\sigma_u = (1 + 0)\,(2.5)\,(1.2(29)\,(10)^6(0.002))$$
$$= 174{,}000 \text{ psi}$$

This is the maximum practical value of tube strength beyond which the deflection limitation is exceeded.

From the Lockheed deflection criterion discussed, if leather back-up rings are used and $D = 1.5$ inches,

$$\sigma_u = \frac{(2.5)\,(1.2)\,(29)\,(10)^6\, 0.0075}{1.5}$$
$$= 435{,}000 \text{ psi}$$

If back-up rings are not used,

$$\sigma_u = \frac{(2.5)\,(1.2)\,(29)\,(10)^6\, 0.004}{1.5}$$
$$= 232{,}000 \text{ psi}$$

It is noteworthy that for $D = 2$ inches the resulting values based on this design criterion are $\sigma_u = 326{,}000$ and 174,000 psi respectively. The latter value is the same as that for the general case of $\delta/D = 0.002$-inches per inch.

The fourth design consideration, overall column stability, has been treated in detail by Hoblit[4]. Based on his analysis, it can be shown that in the region of $a/L = 1/2$,

$$n = \frac{2\, I_2}{I_1 + I_2} \qquad (10)$$

to a first approximation. Then

$$P_{cr} = \frac{n\, \pi^2 (EI)_m}{L^2} \qquad (11)$$

The quantity $(EI)_m$ is defined by

$$(EI)_m = \frac{E(I_1 + I_2)}{2} \qquad (12)$$

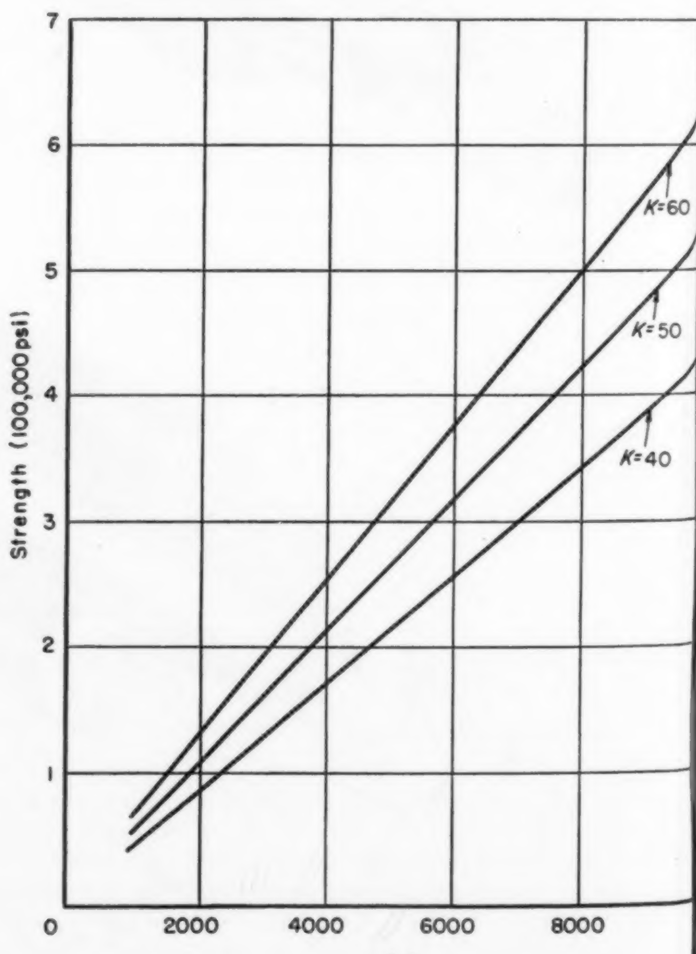


Fig. 4—Chart for determining the practical maximum strength of steel tubes in bending

where I_1 and I_2 are rectangular moments of inertia of the cylinder and rod portions, respectively, of a stepped column such as a hydraulic cylinder. In this case the materials of both rod and cylinder are assumed to be the same.

Combining Equations 10, 11 and 12,

$$P_{cr} = \frac{\pi^2 E I_2}{L^2} \qquad (13)$$

Equation 13 can also be expressed as

$$P_{cr} = B I_2 \qquad (14)$$

where $B = \pi^2 E/L^2$ is constant.

Equation 13 indicates that the buckling load, P_{cr} is not dependent upon the stiffness, or area, of the cylinder tube but rather only on the stiffness of the rod which might be termed the "weakest link." As a result the cylinder tube will seldom be designed by column stability and, therefore, this consideration will have no bearing on the practical maximum strength of the tube. There is of course, the case where the rod diameter does not provide sufficient stiffness. Then, usually, both the cylinder and rod diameters must be increased to meet functional and strength requirements.

The final maximum strength to be used for design is the minimum of the various maximum values determined in the foregoing.

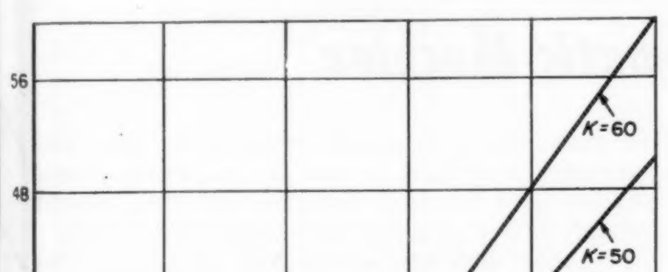


Fig. 5—Chart for determining the practical maximum strength of long tubular columns

Bending Analysis: Maximum strength values may also be derived for other types of loadings, *Fig.* 1. The case of a tube loaded in bending is a typical example. From the standard flexure formula

$$\sigma_b = \frac{MD}{2I} = \frac{MD}{2\left(\frac{\pi}{64}\right)[D^4 - (D-2t)^4]}$$

$$= \frac{32\,MD}{\pi t^4\left[8\left(\frac{D}{t}\right)^3 - 24\left(\frac{D}{t}\right)^2 + 32\left(\frac{D}{t}\right) - 16\right]} \qquad (15)$$

Assuming $\sigma_{ba} = C\sigma_u$ where $C = 1.3$ for large values of D/t, equating $\sigma_b = \sigma_{ba}$ and solving for σ_u gives

$$\sigma_u = \frac{32\,MD}{C\pi t^4\left[8\left(\frac{D}{t}\right)^3 - 24\left(\frac{D}{t}\right)^2 + 32\left(\frac{D}{t}\right) - 16\right]} \qquad (16)$$

Also, assuming for the machinability limit, $K = (D/t)_{max}$, and substituting in Equation 16,

$$\sigma_u = \frac{32K^4M}{C\pi D^3(8K^3 - 24K^2 + 32K - 16)} \qquad (17)$$

Then, for $C = 1.3$,

$$\sigma_u = \frac{7.83\,K^4M}{D^3(8K^3 - 24K^2 + 32K - 16)} \qquad (18)$$

As K becomes large σ_u may be expressed approximately by

$$\sigma_u = \frac{KM}{D^3} \qquad (19)$$

From Equation 18 with $K = 50$, $\sigma_u = 51.8\ M/D^3$. If $M = 10^6$ and $D = 6$, $\sigma_u = 51.8\ (10)^6/216 = 240{,}000$ psi. The approximate formula, Equation 19, yields $\sigma_u = 231{,}000$ psi. If $M = 10^5$ and $D = 3.5$, $\sigma_u = 51.8\ (10)^5/(3.5)^3 = 121{,}000$ psi.

The value of C used in this analysis may readily be derived. By assuming $\sigma_b = C\sigma_u$ and stress distribution is fully plastic, it can be shown that if $D/t = K = 50$, $C = 1.30$. Also if $D/t = K = \infty$, $C = 1.27$. The relationship, $\sigma_b = C\sigma_u$, is theoretically sound for bending modulus of round tubes for some ranges of D/t and is approximately correct for other practical ranges.

A useful graph in which M/D^3 is the abscissa, σ_u is the ordinate, and K is the family parameter is shown in *Fig.* 4.

As an example, the minimum diameter of a shock strut is usually determined by the maximum allowable static pressure. Then, with the diametral bending moment and machinability limit given, the maxi-

mum strength can be determined. Conversely, with a value for strength given, the diameter can be determined. In modern high-speed thin-wing airplanes, the maximum strut diameter would be limited by wing thickness and very high strengths would apply.

Tubular Columns: In the case of long tubular columns the maximum strength criteria may also be readily derived. The applied stress is

$$\sigma = \frac{P}{A} \qquad (20)$$

Allowable load can be expressed as

$$P_{cr} = \frac{C\pi^2 EI}{L^2} \qquad (21)$$

Then, the critical stress is

$$\sigma_{cr} = \frac{P_{cr}}{A} = \frac{C\pi^2 EI}{L^2 A} \qquad (22)$$

The moment of inertia is

$$I = \frac{\pi}{64}\left[D^4 - (D - 2t)^4\right] \qquad (23)$$

By neglecting powers of t greater than unity and substituting $A = \pi Dt$, Equation 23 becomes $I = AD^2/8$. Combining this simplified version with Equation 22 gives

$$\sigma_{cr} = \frac{\pi^2 CED^2}{8L^2} \qquad (24)$$

Using a zero margin of safety and combining Equations 2, 20 and 24 gives

$$E = \frac{8\, l^2\, PK}{C\, \pi^3\, D^4} \qquad (25)$$

This equation has been evaluated in *Fig.* 5 using K as a family parameter.

In the case of columns, E is the measure of strength and is the only factor for identifying the material. Since by Equation 25, E is proportional to K, the equation relates maximum practical strength to the limit for machining.

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Double-End Automatic Machine

CONSISTING of two double-end machine sections placed side-by-side, a new automatic machine permits the operator to set up the machine for an operation on two opposite ends of a cross-shaped workpiece while an operation is being performed on the other two members of the cross. The part to be machined is manually loaded and clamped in a fixture and the fixture is loaded into one machine section and power clamped in position for the initial operations. The fixture is transferred to the second machine section for processing of the remaining members.

Each unit of this machine, developed by Snyder Tool & Engineering Co., is controlled separately, and the work cycle is automatic. Pressing the start button causes the motors to start and the head to rapid traverse until the feed switch is contacted, causing the unit to feed to depth and automatically return.

Feed start dogs and positive stop positions for the heads are selected through a rotating dograil and a movable end stop. The spindle noses in the head are provided with a quick change lock for holding the various tools.

One section of the machine is provided with step drilling attachments for drilling deep holes. Each section has its own hydraulic tank and pump and hand-operated centralized lubrication system.

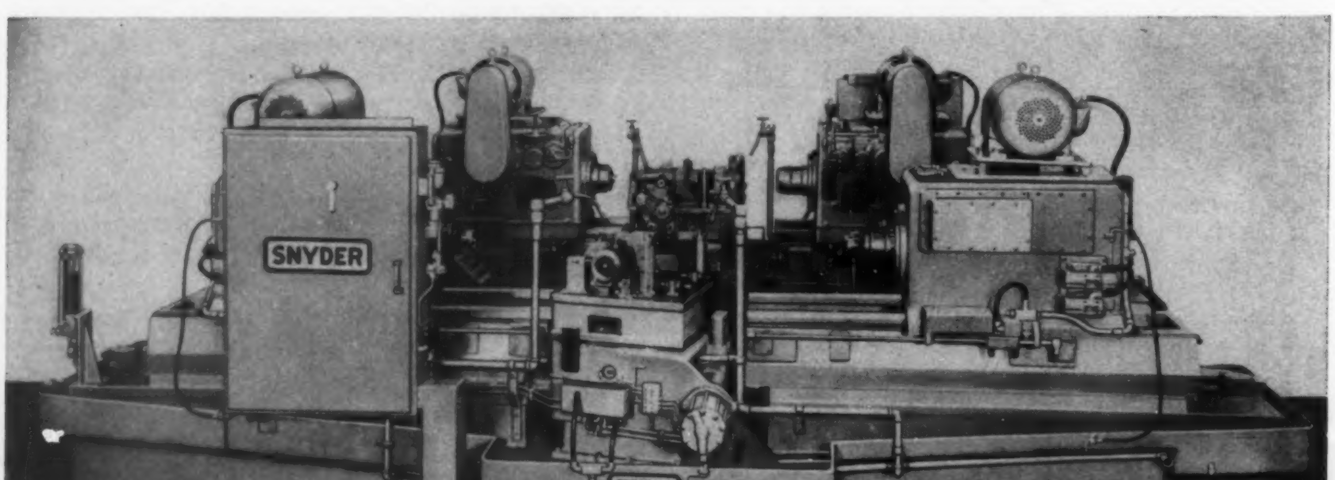
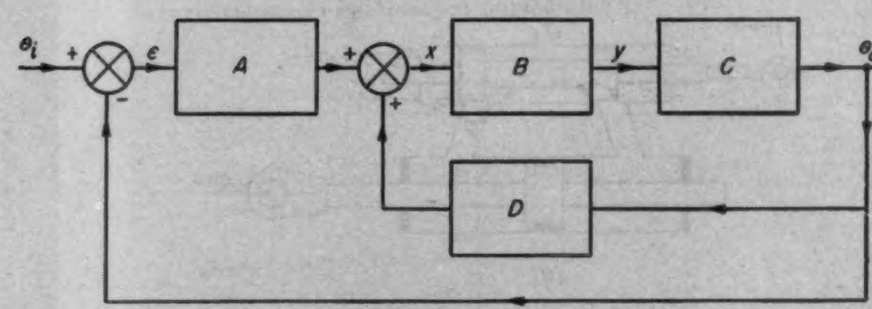


Fig. 1—Block diagram representing a linear servo system which obeys the laws of linear superposition



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ANALYZING NONLINEAR SERVOS

. . . by "linearizing" the performance equations

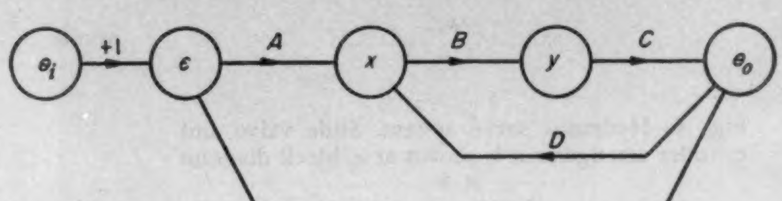


Fig. 2 — Dependence diagram for the system illustrated in *Fig.* 1

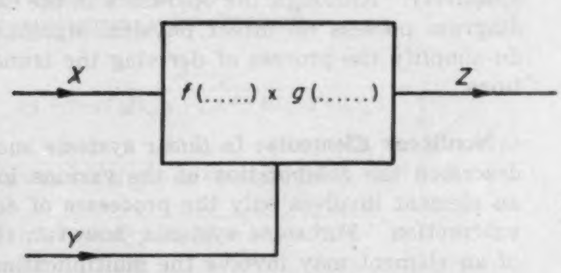


Fig. 3 Nonlinear element for which the output may involve the multiplication or division of two input quantities

POWERFUL methods have been developed for analyzing linear servo systems—systems in which the variables are related by linear differential equations, and which obey the laws of linear superposition. Such a system is generally represented pictorially by a block diagram, *Fig.* 1. Each box represents a physical element which is characterized by a transfer function A, B, C . . . relating the output of the element to its input. A, B, C . . . will normally be algebraic functions of the Laplace operator s, and represent the inertia, damping and resilience and so on within the element.

However, if a particular element has more than one input, its output will in general be a linear function of the several inputs, and the element can no longer be represented by a single transfer function. In this case it may be more convenient to draw a dependence diagram relating the system variables. The variables are placed in boxes while the function operators are written on the connecting lines, and each variable is equated to the algebraic sum of the incoming quantities. The dependence diagram for the system illustrated in *Fig.* 1 is shown in *Fig.* 2.

The equations can be written immediately as

$$\left.\begin{aligned} \epsilon &= \theta_i - \theta_o \\ x &= A\epsilon + D\theta_o \\ y &= Bx \\ \theta_o &= Cy \end{aligned}\right\} \qquad (1)$$

These equations may be compounded to give the

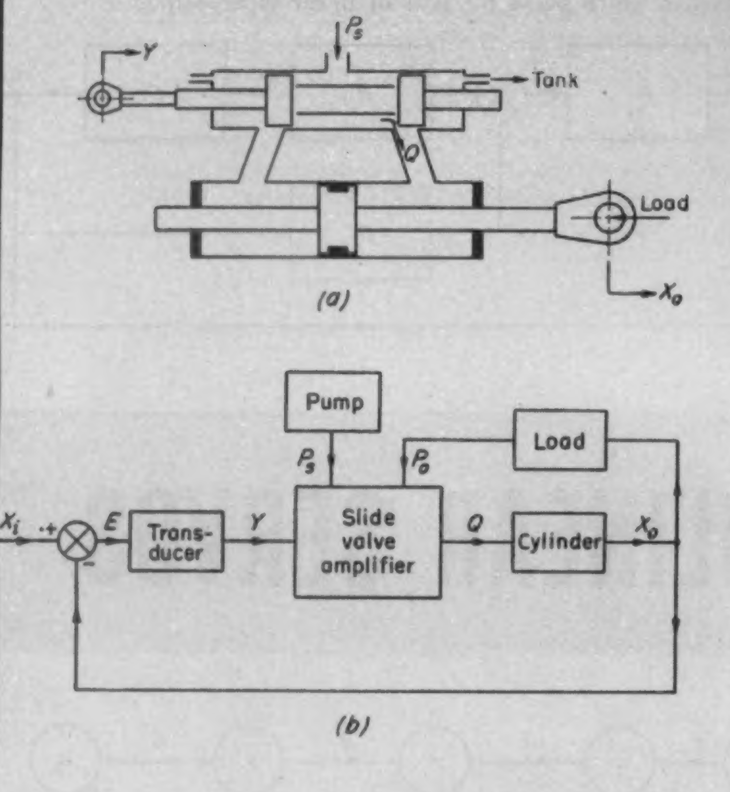


Fig. 4—Hydraulic servo system. Slide valve and cylinder arrangement is shown at *a*, block diagram at *b*

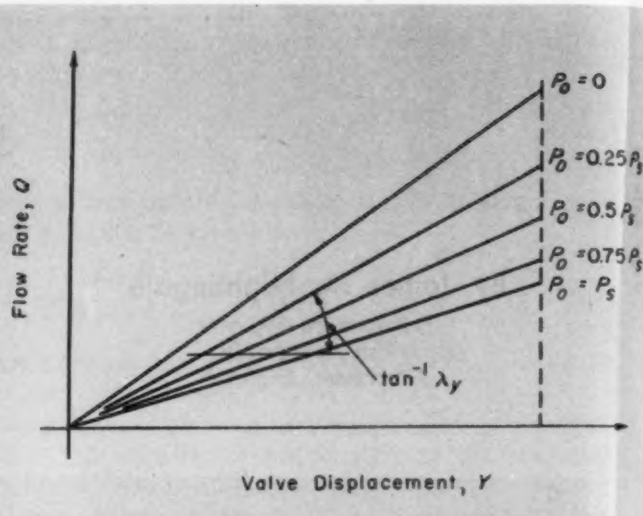


Fig. 5—Curves of flow versus valve displacement and flow versus cylinder operating pressure for slide valve

loop and overall transfer functions θ_o/ϵ and θ_o/θ_i, respectively. Although the operators in the dependence diagram possess no direct physical significance they do simplify the process of deriving the transfer functions.

Nonlinear Elements: In linear systems such as that described the combination of the various inputs into an element involves only the processes of addition or subtraction. For some systems, however, the output of an element may involve the multiplication or division of two or more input quantities. For example, the output Z of the element shown in *Fig.* 3 may be given by an expression of the form

$$Z = f(X) \times g(Y) \qquad (2)$$

The differential equations of a system including such elements are usually intractable, and a study of operating characteristics would involve the solution of nonlinear equations, for which no general methods of investigation have been formulated. The frequency response approach, so powerful in linear servo analysis, cannot be applied directly because the characteristics of the system vary with the amplitude of oscillation. Further, the output will not usually be sinusoidal in wave form.

Frequency response offers such an attractive method of analysis that an attempt to "linearize" a nonlinear system is often well worthwhile. This may be done by considering small motions about steady-state conditions. Then separating perturbations from steady-state quantities permits the formulation of a series of linear equations each of which defines the performance within a narrow band about a particular steady-state position.

Slide Valve Hydraulic Servo: Probably the best way to demonstrate this technique is by an actual example, and the system chosen is a hydraulic servo in which the main amplifier is a slide valve, *Fig.* 4.

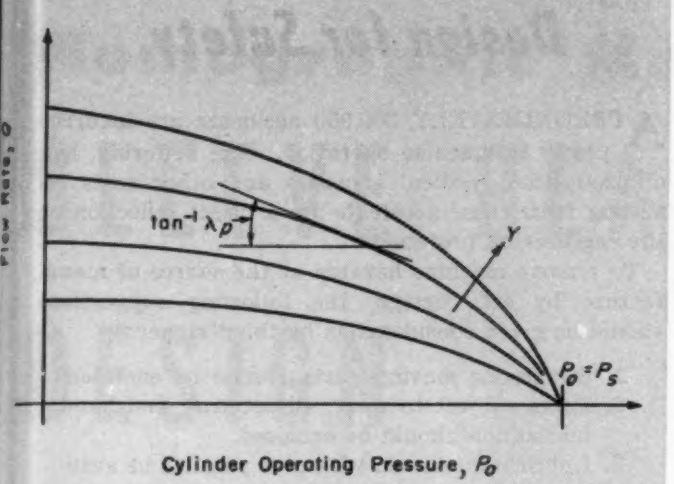
The power source is a pump which delivers fluid at a constant pressure P_s, this flow being metered by a slide valve to the servo motor, which in this case is a hydraulic cylinder. The valve constitutes a variable orifice, and since an orifice is an energy dissipation element there will be a pressure drop across the valve. The flow Q through the valve is proportional to the product of valve displacement Y and the square root of the pressure drop $P_s - P_o$ across the valve, and since the latter depends on the output X_o a nonlinearity of the type given by Equation 2 is introduced.

In the following list of the variables, capital letters denote steady-state quantities, small letters denote small increments:

X_i, x_i = Input
X_o, x_o = Output
Q, q = Flow
Y, y = Valve displacement
P_o, p_o = Cylinder operating pressure
E, ϵ = Error signal

Following are the system parameters:

P_s = System pressure
A = Cylinder piston area
λ = Transducer gain
T = Time constant of valve
K = Valve flow constant



λ_y, λ_p = Flow coefficients
M = Output inertia
b = Output damping coefficient
k = Output spring stiffness

The elements taken in order round the loop, *Fig.* 4*b*, are the error sensing device, a transducer, slide valve amplifier, and the hydraulic cylinder. The transducer converts the error signal into a valve displacement, probably with some time delay. The steady-state system equations are

$$\left.\begin{aligned} E &= X_i - X_o \\ Y + T\frac{dY}{dt} &= \lambda E \\ Q &= K\,Y\sqrt{P_s - P_o} \\ X_o &= \frac{1}{A}\int Q\,dt \\ AP_o &= M\frac{d^2X_o}{dt^2} + b\frac{dX_o}{dt} + kX_o \end{aligned}\right\} \qquad (3)$$

so that the following transformed relations for the

ANALYZING NONLINEAR SERVOS

perturbations can be obtained immediately:

$$\left.\begin{aligned} \varepsilon &= x_i - x_o \\ \frac{y}{\varepsilon} &= \frac{\lambda}{1+Ts} \\ \frac{x_o}{q} &= \frac{1}{As} \\ \frac{p_o}{x_o} &= \frac{1}{A}(Ms^2 + bs + k) \end{aligned}\right\} \qquad (4)$$

The system can be linearized if flow can be expressed in the form $Q = Ay + Bp_o$. For small increments in Q, Y and P_o

$$Q + q = K(Y + y)\sqrt{P_s - (P_o + p_o)}$$

Expanding gives

$$Q + q = KY\sqrt{P_s - P_o} + \frac{\partial Q}{\partial Y}y + \frac{\partial Q}{\partial P_o}p_o + \cdots$$

Thus the perturbed flow may be written

$$q = \lambda_y y - \lambda_p p_o \qquad (5)$$

where

$$\lambda_y = \frac{\partial Q}{\partial Y} = K\sqrt{P_s - P_o}$$

$$\lambda_p = -\frac{\partial Q}{\partial P_o} = \frac{KY}{2\sqrt{P_s - P_o}}$$

The flow-displacement and flow-pressure curves in *Fig.* 5 show the physical significance of λ_y and λ_p.

The dependence diagram may now be drawn, *Fig.* 6. From Equations 4 and 5 the open loop and overall transfer functions may be obtained.

$$\left.\begin{aligned} \frac{x_o}{\varepsilon} &= \frac{A\lambda\,\lambda_y}{(1+Ts)[M\lambda_p s + (A^2+b\lambda_p)s + k\lambda_p]} \\ \frac{x_o}{x_i} &= \frac{A\lambda\,\lambda_y}{M\lambda Ts^3 + [M\lambda_p + (A^2+b\lambda_p)T]s^2 + [k\lambda_p T + (A^2+b\lambda_p)]s + k\lambda_p + A\lambda\lambda_p} \end{aligned}\right\} \qquad (6)$$

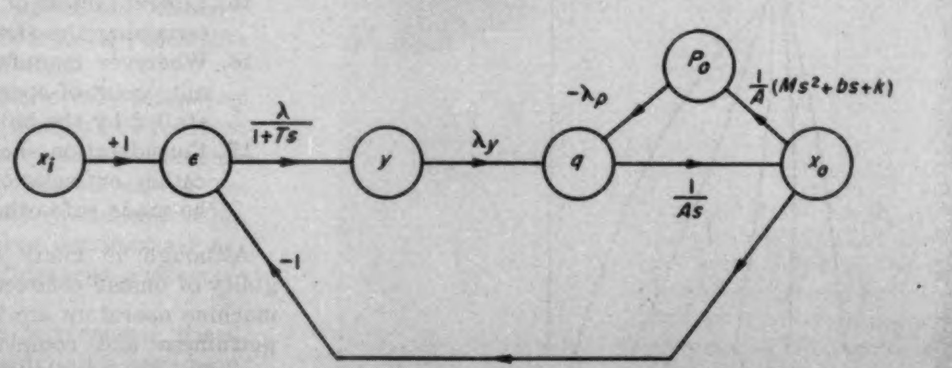


Fig. 6 — Dependence diagram of the hydraulic servo shown in *Fig.* 4

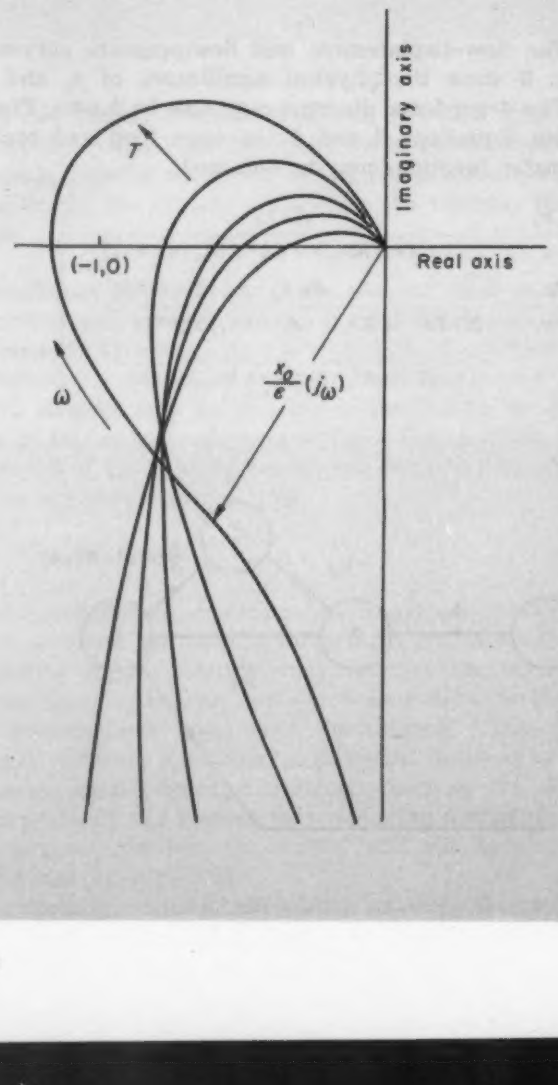
The performance can now be investigated in the neighborhood of each equilibrium position; for example, a series of Nyquist plots may be drawn, *Fig.* 7.

Value of Linearized Analysis: Although the process outlined is only approximate, the results obtained give a qualitative description of the system, and afford useful information about the stability of the system on the threshold of motion.

It would be impossible to consider the motion about every equilibrium position, and tiresome to attempt it. Instead stability may be defined in terms of the steady-state coefficients, and the relations derived may be plotted in graphical form. Prohibitive zones are defined, within which the system becomes unstable.

The fact that a system is unstable at a certain point does not necessarily mean that the oscillations will increase indefinitely, since the stiffness and damping change with amplitude. However, the presence of any continuous oscillation in a servo is usually to be deplored, and the parameters should be adjusted to ensure a stable system throughout its entire operating range.

Fig. 7—A Nyquist plot in the region of an equilibrium position for the hydraulic servo, showing in this instance the effect of the valve time constant T on stability



Design for Safety

APPROXIMATELY 300,000 accidents are incurred yearly in machine operation. The suffering, loss of production, medical expenses and other costs resulting from these accidents are a direct reflection on the engineering profession.

To remove machine hazards at the source of manufacture by safe design, the following suggestions should be given consideration by the designer:

1. Dangerous moving parts should be enclosed.
2. Parts subject to wear, adjustment, and hand lubrication should be enclosed.
3. Lubrication should wherever possible be automatic and continuous when the machine is in operation.
4. Consideration should be given to individual drives so that hazards due to driving mechanisms may be eliminated.
5. Sharp contrast between light and shadow and glare in the vicinity of point of operation should be avoided.
6. Materials should be mechanically conveyed to, and products away from, machines.
7. Provision should be made for automatically conveying dusts and gases away from machines.
8. Noise should be eliminated or reduced.
9. Vibration should be eliminated or reduced as much as possible.
10. Machine motions tiring to the eyes should be avoided.
11. Exterior shapes of any parts of the machines that require frequent contacting or handling should be such as to facilitate convenience in handling.
12. Weight of parts to be handled should be kept within the limits of convenience.
13. Throughout the design of the machine and its parts, consideration should be given to convenience in attaching accessories, chiefly point-of-operation guards.
14. Consideration in design should be given to the external shape of the machine unit so that the danger of accident from tripping and falling, and collision, will be minimized.
15. Liberal factors of safety should be used in determining the strength of parts.
16. Wherever manufacturing circumstances permit, point-of-operation guards should be installed by the builder of the machine.
17. Consideration should be given to the safe location or isolation of machines that cannot be made safe otherwise.

Although in many cases machine operators are guilty of unsafe conduct, engineers should realize that machine operators are human—subject to worry, forgetfulness and complacency. An engineer should recognize the possibility of human failings and strive to make his designs as foolproof as possible.

—*ASME Safety Division*

Nomographic Method for SLEEVE BEARING DESIGN

A simplified procedure for comprehensive consideration of the many variables that enter into the design of medium and large-size sleeve bearings

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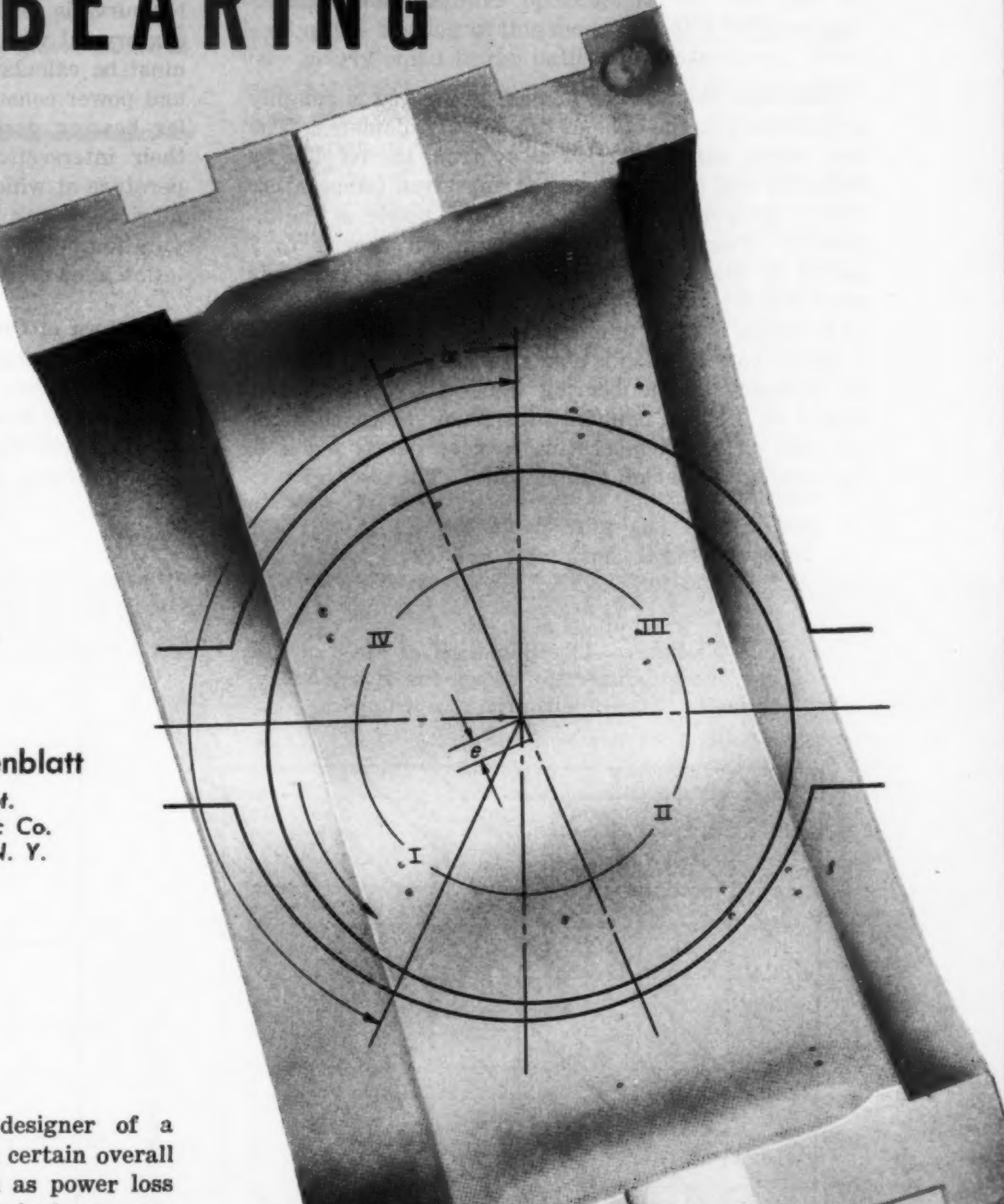


Fig. 1—Basic design approach was worked out on split cylindrical-bore liner-type bearing having two axial oil distribution grooves at the split; a cross section is also shown

THE problem confronting the designer of a sleeve bearing is to predetermine certain overall aspects of its performance, such as power loss and operating temperature, under certain known operating conditions, such as load, speed, and viscosity of lubricant. Dimensions must be chosen which give the safest and most economical design.

Performance of a sleeve bearing is very intimately associated with the rate at which heat can be removed from the bearing oil film. In larger bearings, the bulk of this heat must be removed by the oil leaving the bearing. Consequently, detailed study and knowledge of oil flow characteristics of sleeve bearings aid accurate computation of their performance characteristics.

Methods of computing bearing performance based on knowledge of the oil flow characteristics have been described previously.[1] These methods are summarized

1. References are tabulated at end of article.

here in the form of easy-to-use alignment charts. These charts permit bearing calculations to be made rapidly and to be kept as a permanent record.

Basic Considerations: Power consumption of a sleeve bearing resulting from the shearing of oil in the bearing oil film is given approximately by Petroff's relation

$$W_p = \frac{2\pi^3 ZN^2 D^3 L}{C} \quad (1)$$

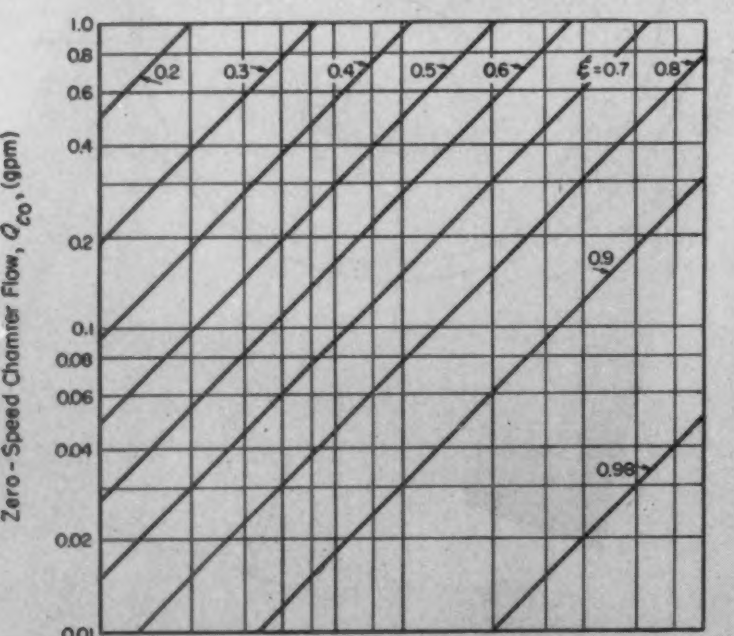
Note that the amount of heat generated is roughly proportional to the cube of the bearing diameter. The heat which can be carried away from the oil film by radiation and by conduction at any given temperature level is proportional only to the first power of the diameter. Therefore, in bearings much over 1 to 2 inches in diameter, oil flowing through the bearing must remove the bulk of the heat generated in order to maintain bearing temperatures at reasonable values.

Power consumption is also a function of the average oil viscosity in the bearing film. Careful measurements of oil film temperatures have shown that the average oil film temperature is very nearly equal to the average outlet oil temperature. The average outlet oil temperature is dependent not only upon the oil flow through the bearing but also upon the power consumption of the bearing.

The design problem in its basic simplicity, therefore, consists in determining, first, the average outlet oil temperature of the bearing as a function of oil viscosity and, second, the viscosity of the oil to be used in the bearing as a function of temperature. The latter curve is readily determined from conventional laboratory oil viscosity measurements, while the former must be calculated from a knowledge of the oil flow and power consumption characteristics of the particular bearing design. When these curves are plotted, their intersection represents the viscosity and temperature at which the given bearing will operate when lubricated with the particular oil chosen for the computation. This is the basis for the method of calculation shown on the accompanying alignment charts.

Pattern of Oil Flow: The assumption that the average oil film temperature is equal to the average outlet oil temperature of a bearing is easier to understand if the entire bearing is considered as a mixer. Under average conditions the end flow *from* the bearing is a small portion of the circumferential or circulating

Fig. 2—Below—Determination of kinetic energy correction factor for flow through the chamfer openings



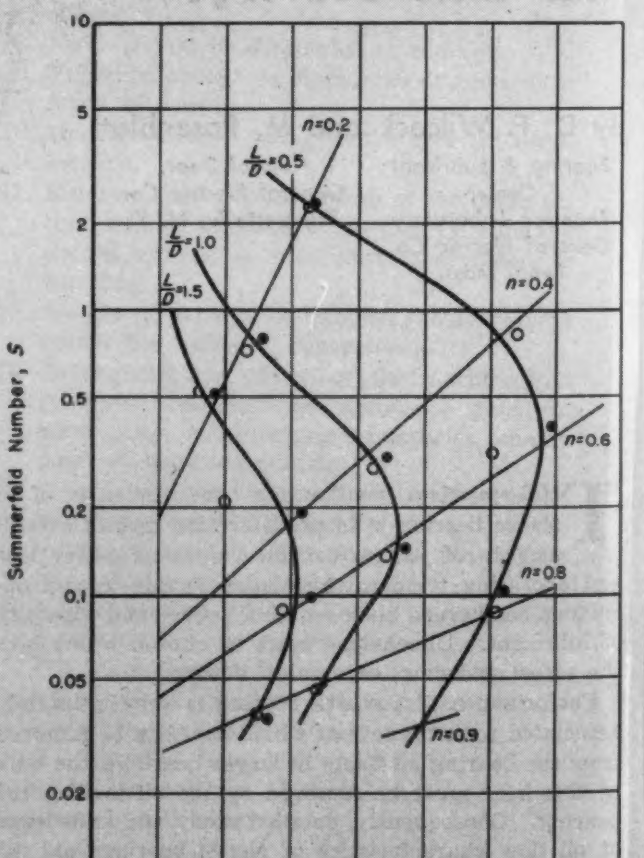


Fig. 3—Right—Dimensionless oil flow coefficient for the bearing as a function of Sommerfeld number for constant length-diameter ratio and constant eccentricity

flow *within* the bearing. Thus, the bearing may be likened to a mixer to which is being added a small stream of cold oil and from which is being withdrawn continuously a small stream of oil at the average temperature of the mixer. The mixer temperature is equal to the temperature of the oil being withdrawn but is above the temperature of the oil being added by an amount sufficient to carry away the heat being generated within the mixer.

Bearings of the split type normally have chamfers at the ends of the oil grooves for two purposes, to break the sharp corners, and to flush some additional oil through the bearing. Careful experimental measurement of the temperature of the small oil streams leaving these chamfer openings shows that the oil leaving the bearing in this manner is at a temperature very nearly equal to the outlet oil temperature. This is rather direct evidence of the complete mixing which takes place in the oil feed grooves and of the mixer-type action of oil flow in sleeve bearings.

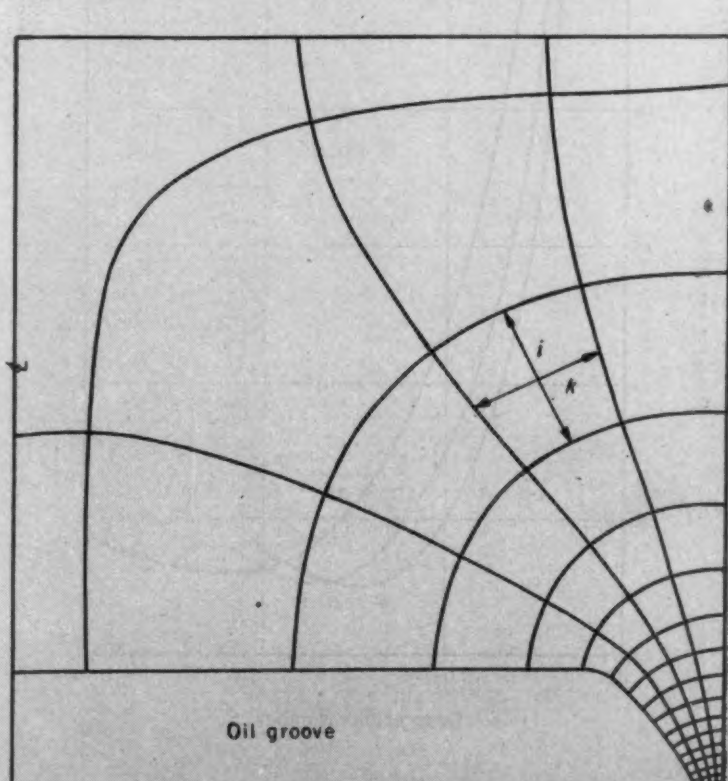
The type of bearing for which this analysis was originally developed is illustrated in *Fig.* 1. This bearing is split into two halves. It has a cylindrical bore and has large axial oil distribution grooves located at the split. Each groove subtends an angle of approximately 30 degrees between blends with the cylindrical bore. While the analysis was developed for this particular type of bearing, it is general enough in principle so that the reader can with slight modification apply it to other designs of pressure-fed bearings.

Three types of oil flow take place in bearings of this type, (1) end flow from the loaded half of the bearing, (2) end flow from both halves of the bearing oil film due to the inlet oil pressure in the

Nomenclature

- C = Diametral clearance, inches
- C_p = Heat capacity of oil, Btu per gal per deg C
- D = Journal diameter, inches
- G = Total bearing load, lb
- H = Power consumption of bearing, hp
- L = Axial length of bearing, inches
- N = Journal speed, rpm
- P = Bearing load per unit of projected area, psi
- Q = Total oil flow, gpm
- Q_c = Oil flow through single oil groove chamfer, gpm
- Q_F = Oil flow through bearing oil film due to inlet pressure, gpm
- Q_R = Oil flow due to rotation of journal, gpm
- R = Journal radius, inches
- S = Sommerfeld number $= (D/C^2\ (Z_2N/P) = 2.42 \times 10^{-9}\ (D/C)^2\ (\mu_2N/P)$
- T = Temperature, deg C
- ΔT = Temperature rise of oil, $T_2 - T_1$, deg C
- W = Power loss transferred to shaft, lb-in. per minute
- W_p = Power loss with shaft centered (Petroff's equation value), lb-in. per minute
- Z = Oil viscosity, lb-minute per in.^{2}
- a = Chamfer dimension radial to journal, inches
- b = Chamfer dimension tangent to journal, inches
- e = Journal eccentricity, inches
- h = Oil film thickness, inches
- j = Dimensionless power loss ratio $= W/W_p$
- l = Axial length of chamfer, inches
- m = Number of chamfers
- n = Eccentricity ratio $= 2e/C$
- p = Oil pressure, psi
- q = Dimensionless oil flow coefficient $= 294\,Q_R/NDLC$
- t = Slot thickness, inches
- v = Film flow coefficient determined from flux plot
- w = Slot width, inches
- α = Acute angle between load vector and line of centers
- μ = Oil viscosity, centipoises $= 4.14 \times 10^8 Z$
- ξ = Ratio of corrected to uncorrected chamfer flow (kinetic energy correction) $= Q_c/Q_{ctot}$
- 0 = Subscript, zero speed
- 1 = Subscript, at oil inlet
- 2 = Subscript, at oil outlet
- min = Subscript, at point of minimum approach
- tot = Subscript, total

Fig. 4—Method of solution for film flow coefficient by means of a flux plot in the region of the oil inlet, showing one-eighth of total bearing area



oil grooves, and (3) flow through the chamfers at the ends of the oil grooves. It has been found possible to analyze each of these flows separately and so to compute the total oil flow for any given bearing under known operating conditions. The total oil flow is given by

$$Q = Q_F + Q_{ctot} + Q_R = \frac{1.87 \times 10^4 \, v \, C^3 \, p_1}{\mu_2} +$$

$$\xi \, m \, \frac{47{,}200 \left(a + \dfrac{C}{2} \right)^4 p_1}{l \, \mu_2} + q \, \frac{NDLC}{294} \quad \ldots\ldots \quad (2)$$

In this equation ξ is determined from *Fig.* 2; q is determined from *Fig.* 3; and m is the number of chamfered openings, usually twice the number of oil grooves. Details of this analysis are contained in the previous reference.

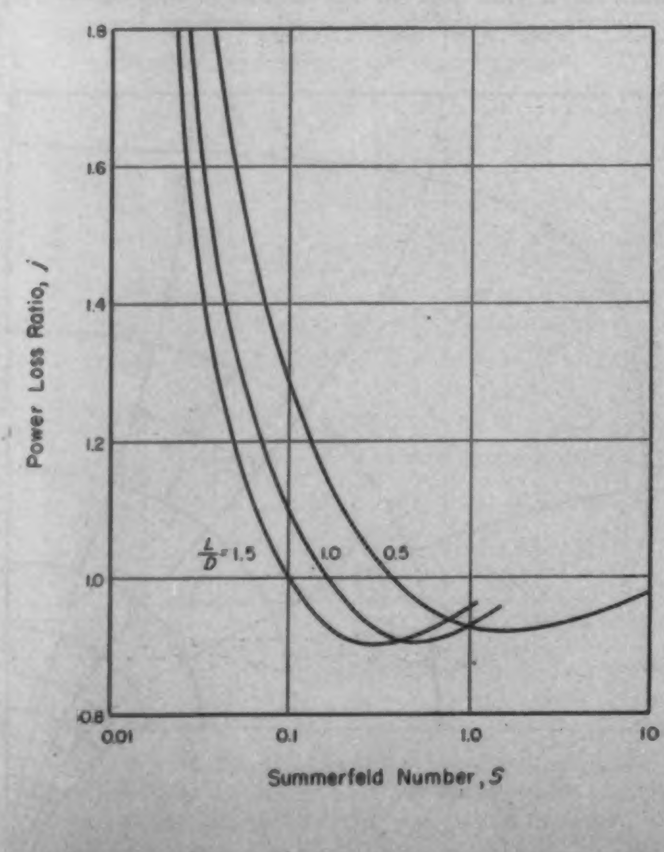
The dimensionless coefficient v is based on the oil groove configuration. It is the equivalent width to length ratio of a thin slot, having the same thickness as the bearing oil film, which would carry the same oil flow as the film in the neighborhood of the oil inlet grooves. An example of a "flux plot" to determine this coefficient is shown in *Fig.* 4, and the method of making this plot is outlined later. Note that only one-eighth of the flow is shown in this diagram. The coefficient for this groove configuration is $v = 8 \times 4.05 = 32.4$.

Both chamfer and film flow expressions assume that the approximate film thickness at the oil inlet groove is $C/2$. For more accurate work the film flow term, Q_F, may be multiplied by

$$1 + 12 \left(\frac{e \sin \alpha}{C} \right)^2$$

to correct for shaft eccentricity e. If there is an oil inlet at some other position than the horizontal, its film flow term may be multiplied by the quantity $(2h/C)^3$ where h is the film thickness at the oil inlet. The chamfer flow term may likewise be corrected by substituting the film thickness at the chamfer for the quantity $C/2$ in the chamfer flow expression.

Fig. 5 — Power loss ratio for three values of constant length-diameter ratio



Computation of Outlet Oil Temperature: Outlet oil temperature is computed as a function of two or more assumed oil viscosities, and the resulting curve is plotted on viscosity versus temperature co-ordinates in order to obtain an intersection with the viscosity-temperature curve for the lubricating oil. The outlet oil temperature may be calculated from

$$T_2 = T_1 + \frac{42.4 \, H}{C_p Q} = T_1 + f\,(\mu_2) \quad \ldots\ldots \quad (3)$$

Note that both power loss H and oil flow Q are functions of oil viscosity. Equation 3 may be rewritten as

$$T_2 = T_1 + 1.60 \times 10^{-11} \left(\frac{jLD^3N^2}{CQC_p} \right) \mu_2 \quad \ldots\ldots \quad (4)$$

where oil flow Q is determined from Equation 2, and j (determined from *Fig.* 5) is a dimensionless power consumption coefficient which gives the ratio of the true power consumption to that computed from Equation 1.

Equation 4 may be solved numerically for any given combination of the operating variables and bearing dimensions, and the results plotted graphically on the viscosity-temperature chart for the lubricating oil to give a graphical solution for the outlet oil temperature of the bearing. Once this outlet oil temperature has been determined, values for oil flow, power loss, film thickness, etc., may be computed readily.

Use of Alignment Chart: In order to simplify calculations and to give a better physical picture of how the various factors enter into bearing performance, the alignment charts of *Fig.* 6 have been prepared. These charts are arranged to determine: *6a*, Sommerfeld number and the rotational oil flow; *6b*, film oil flow due to inlet pressure; *6c*, chamfer oil flow; and *6d*, power loss and temperature rise of the oil flowing through the bearing. Also included as a part of *Fig. 6d* is a viscosity-temperature chart showing typical curves for light, medium and heavy turbine oils upon which the operating curve for the bearing, as determined from the charts or from Equa-

tion 4, may be plotted. Finally, a nomograph for determining minimum oil film thickness, *Fig.* 6*e*, is included.

The chart is quite general except for the dimensionless q versus S plot of *Fig.* 6*a*. This particular plot, based on similarity laws, holds only for 150-degree bearings; similar plots of q versus S obtained from calculations or experience, however, may be substituted for other bearing configurations. For example, *Fig.* 7 illustrates q versus S charts for 120, 150, and 360-degree bearings the latter consisting of two 180-degree halves without an axial oil distribution groove.[2, 3]

In nomograph b of *Fig.* 6, film flow coefficient v is determined from a flux plot or field map of the oil flow in the region of the oil inlet or chamfer, as shown in *Fig.* 4. Trial and error sketching can be used to determine this pattern. Lines from the oil groove to the outer edge of the bearing are lines of constant flow; intersecting lines are of constant pressure. Usual procedure is to sketch in the latter freehand. The intersecting lines are then drawn so that $i = k$, (giving curvilinear squares) and all intersections are perpendicular, including intersections with the boundaries of the flow "field." Coefficient v then equals the number of flow increments divided by the number of pressure increments for the complete bearing. One point to watch is that intersections must be kept perpendicular (no fudging allowed) so that the pattern will be representative. A good procedure is to use tracing-paper overlays over preceding sketches until the correct pattern is obtained; further details of the procedure can be found in most electrical engineering handbooks.

Minimum oil film thickness, determined with nomograph 6*e*, is dependent only on the diametral clearance C and shaft eccentricity ratio n. The value of n to be used here may be read from the q versus S plot in *Fig.* 6*a*. Curves of constant n are shown so that n may be determined from the Sommerfeld number and the L/D ratio for the bearing.

Alignment Chart Applications: Many varied types of bearing design problems can be worked with the aid of the alignment chart. Since the basic method of the nomograph procedure is shown in *Fig.* 6, results will only be indicated in order to show the utility of the method.

Example 1 — Determining Operating Temperatures: This example illustrates how increasing the chamfer size results in a lower operating temperature for the bearing. The bearing is 8 inches in diameter, 4 inches long, and has a diametral clearance of 0.0126-inch. It is to be operated at 2700 rpm under 150 psi average bearing load. The inlet oil temperature is 40 C and the inlet oil pressure, 10 psi. Chamfer (radial dimension of chamfer, a) is 0.015-inch, and the chamfer is 0.263-inch long. Film-flow coefficient v is 32.4, as previously determined. The operating line is determined for assumed viscosities of 15 and 25 centipoises. The computed temperature rise as shown on *Fig.* 8 is 15 C for 15 centipoises assumed viscosity and 26 C for 25 centipoises assumed viscosity. The resultant operating line is shown as curve 1 on the viscosity-temperature plot. Note that in general the operating line is a curve and the computation of a third point on this line would serve to define the curve and to give a somewhat more accurate determination.

Now that the operating line has been determined, it becomes immediately apparent that on a light turbine oil the bearing will operate with an average outlet oil temperature of approximately 55 C. When using a medium turbine oil, the outlet oil temperature will be approximately 63 C; and with a heavy turbine oil the outlet oil temperature will be about 66 C. From the oil viscosities corresponding to these temperatures for these three oils, the other characteristics of the bearing such as oil flow and power loss can be computed quickly with the alignment chart.

Example 2 — Increasing Chamfer Size: If the chamfer size is now increased fourfold from 0.015-inch to 0.060-inch, with the same length and the same assumed viscosities of 15 and 25 centipoises, the resultant temperature rises are 9.6 and 18.8 C giving operating line 2 on the viscosity-temperature chart. This indicates average outlet oil temperatures of

(*Text continued on Page* 155)

Typical Design Results

Table 1

	Curve 1 (C)	Curve 3 (C)	Difference (C)
Inlet temperature	40	60	20
Operating temperature			
Light oil	55	67	12
Medium oil	63	74	11
Heavy oil	66	76	10

Table 2

Bearing Clearance (in.)	Outlet Oil Viscosity (centipoise)	Outlet Oil Temperature (C)	Oil Flow (gpm)	Power Consumption (hp)
0.008	13.6	73.9	1.85	9.0
0.0126	21.2	61.0	3.11	9.6
0.020	30.6	52.0	5.47	9.6

Table 3

L/D	T_2 (C)	Q (gpm)	H (hp)	H/L (hp/in.)	h_{min} (in.)
0.5	61	3.1	9.6	2.4	0.0027
1.0	66	3.8	14.5	1.8	0.0037
1.5	72	3.7	17.7	1.5	0.0043

Table 4

Oil	D (in.)	L (in.)	P (psi)	μ_2 (centipoise)	T_2 (C)
Light	6.30	9.45	675	10.0	67
Medium	5.75	8.62	810	14.0	75
Heavy	5.66	8.49	835	15.1	78

Fig. 6—Nomographs for Sleeve Bearing Design

The following nomographs are based on analysis of total oil flow through a proposed sleeve bearing design; the total oil flow consists of (see Nomenclature or work sheet)

$$Q = Q_R + Q_F + Q_{ctot}$$

Three nomographs, 6*a* through 6*c*, give the solution for the total flow, as follows:

$$Q_R = \frac{qCDLN}{294} \quad \text{Fig. 6}a$$

$$Q_F = \frac{1.87 \times 10^4 \, v C^3 p_1}{\mu_2} \quad \text{Fig. 6}b$$

$$Q_{ctot} = \frac{47{,}200 \, \xi \, p_1}{l \, \mu_2} \left(a + \frac{C}{2} \right)^4 m \quad \text{Fig. 6}c$$

After total flow is determined, the temperature rise in the bearing may be computed from nomograph 6*d*, which is based on the equations,

$$H = \frac{2.42 \times 10^{-9} \pi^3 j N^2 D^3 L \mu_2}{198{,}000 \, C} \quad \text{Fig. 6}d$$

$$\Delta T = \frac{42.4 \, H}{Q \, C_p} \quad \text{Fig. 6}d$$

By variation of any one of several factors, a series of points representing outlet oil temperatures is obtained, which can be plotted on the graph of nomograph 6*d* to determine an "operating line." Intersection of the curve drawn through these points with a viscosity-temperature curve for the oil used determines the final conditions.

Also useful are two other equations.

$$h_{min} = \frac{C}{2} (1 - n) \quad \text{Fig. 6}e$$

$$S = \frac{2.42 \times 10^{-9} \mu_2 N}{P} \left(\frac{D}{C} \right)^2$$

A work sheet is also included, on which the basic working dimensions and other conditions may be entered.

Methods of Solution

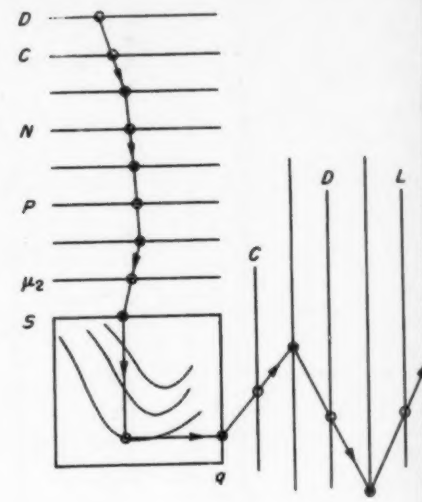
General Method: The dimensions and conditions which are known are: (*a*) D, C, N, P, L; (*b*) same, plus p_1; (*c*) same, plus l, a, m; (*d*) same, plus C_p, T_1. Film-flow coefficient v is determined from a flux plot (see text); j is determined from the q versus S plot on nomograph 6*a*. Viscosity μ_2 (μ on nomograph 6*d*) is the independent variable; values of μ_2 are assumed to obtain the "operating line." If no chamfers are to be used, nomograph 6*c* is not used. This method of solution is outlined in the sketches.

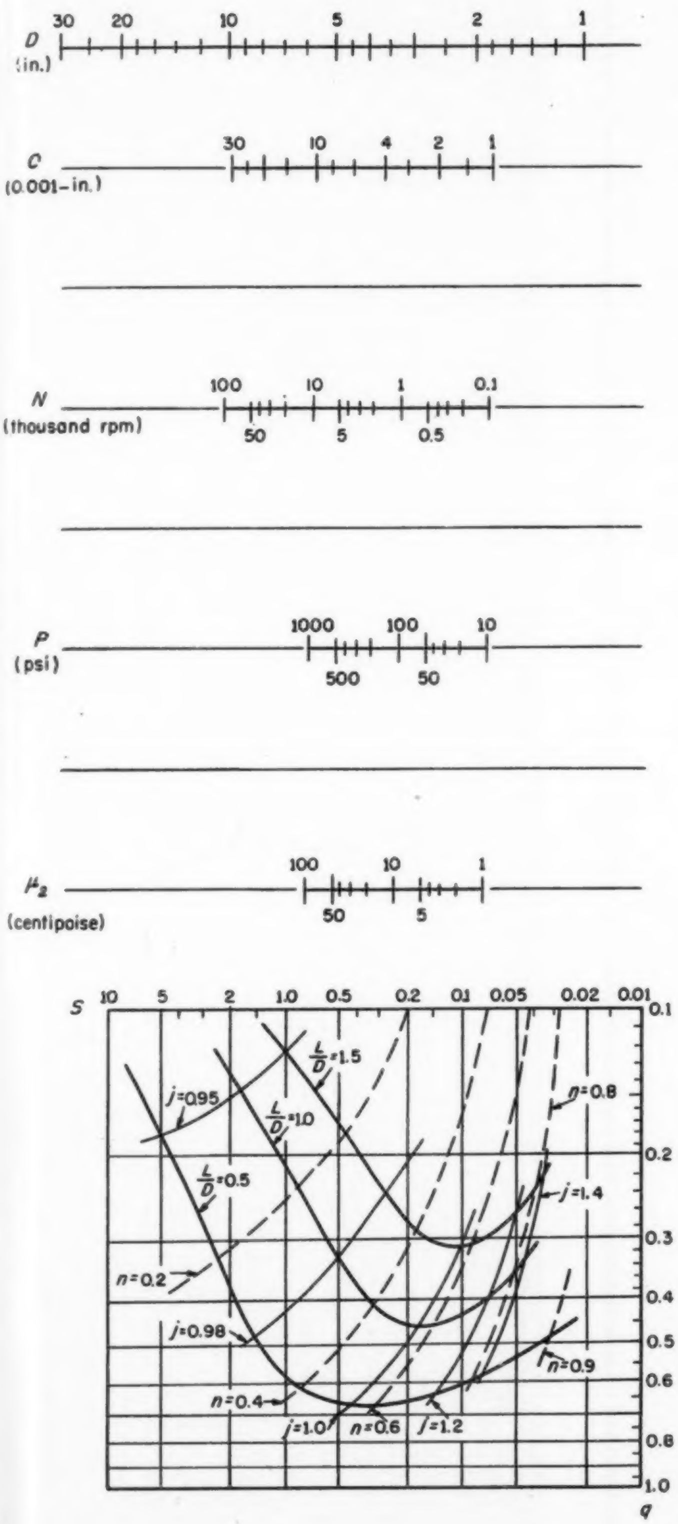
Clearance for Fixed Temperature Rise: Assumed dimensions and conditions are the same as in the general method except that ΔT is also known, permitting μ_2 to be determined from the ΔT-T_1-T_2 section and graph of nomograph 6*d*. The independent variable is C, and various clearance values are assumed until the correct clearance is determined on a trial-and-error basis, shown by correspondence of ΔT in nomograph 6*d* with the desired temperature rise. A first approximation to the clearance may be obtained from

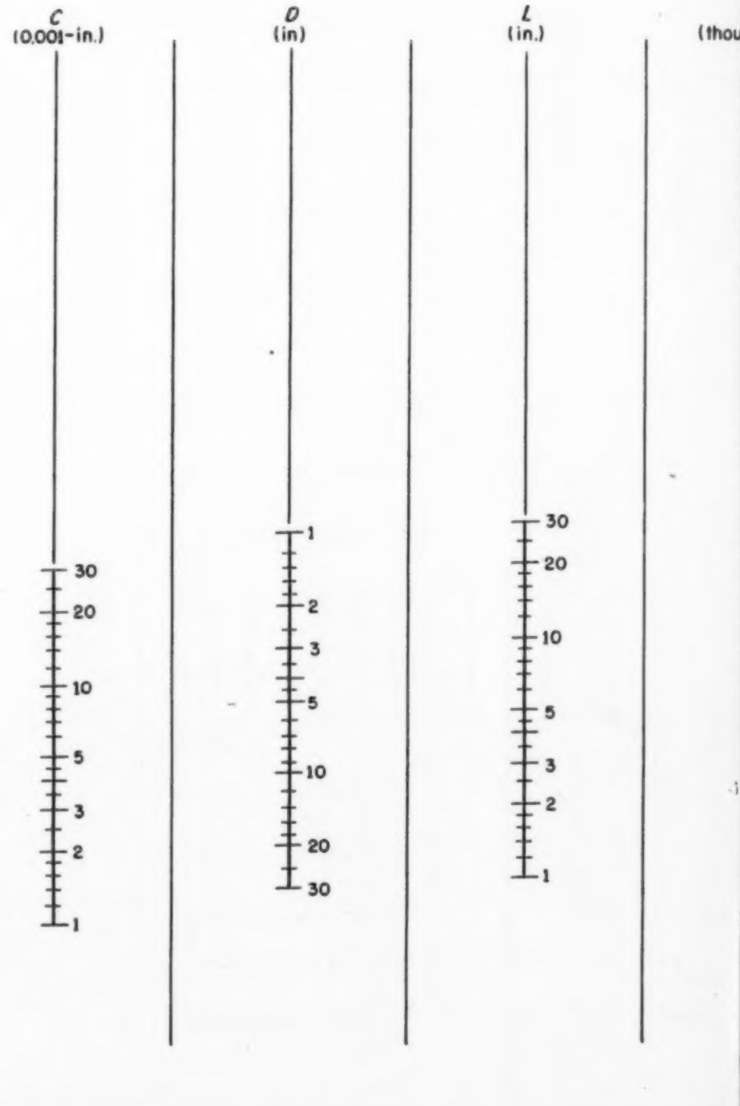
$$C^2 = \frac{1.03 \times 10^{-9} \mu_1 D^2 N}{p_1} + \left(-1 + \sqrt{1 + \frac{25.8 p_1}{C_p(\Delta T)} \frac{\mu_2}{\mu_1} \frac{L}{D}} \right)$$

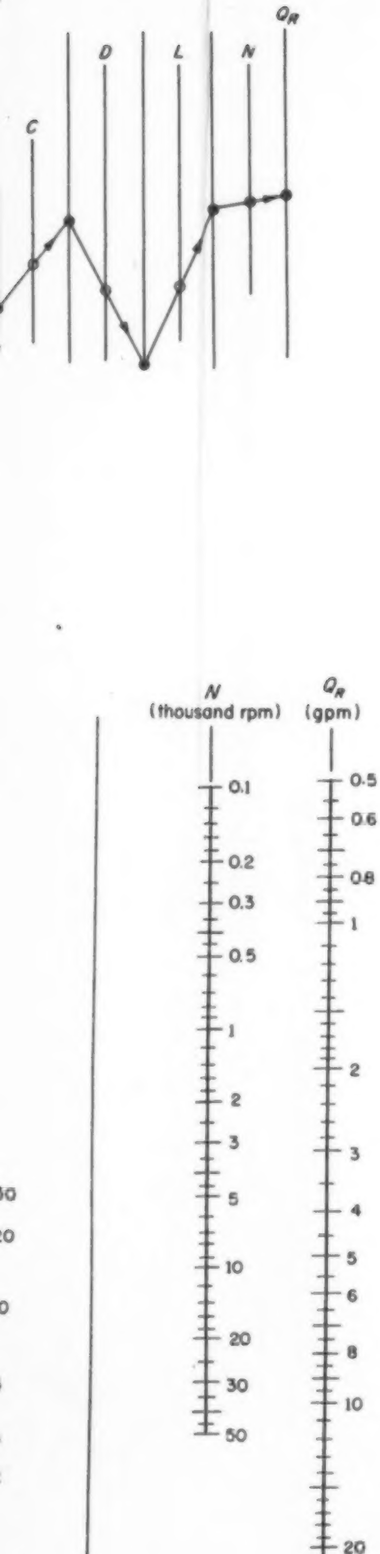
Size for Safe Oil Film Thickness: A trial procedure requiring some judgment is necessary for this determination. Thus, N, G, v, p_1 and h_{min} are known. Initially, chamfer flow is assumed to be zero, eliminating the need for nomograph 6*c*. A value for C is assumed which, by use of nomograph 6*e*, fixes n. By assuming an L/D ratio, values for S and q can be determined from the q versus S plot of nomograph 6*a*. Further assumption of several values for D fixes L and the unit loading P for each of these values. For each value of D, a value of μ_2 can be found by using nomograph 6*a* north of the q versus S plot; going east of this plot, Q_R can be determined. The solution is then straightforward, with one value of Q_R, Q_F, Q and ΔT being found for each value of D; the several points determine the "operating line." Using the intersection of this line with the curve for the oil to be used, the solution is then worked "backward" to determine the bearing design. A similar method can be used for varying L instead of D.

(a)—Rotational Flow

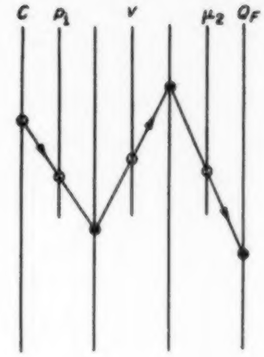


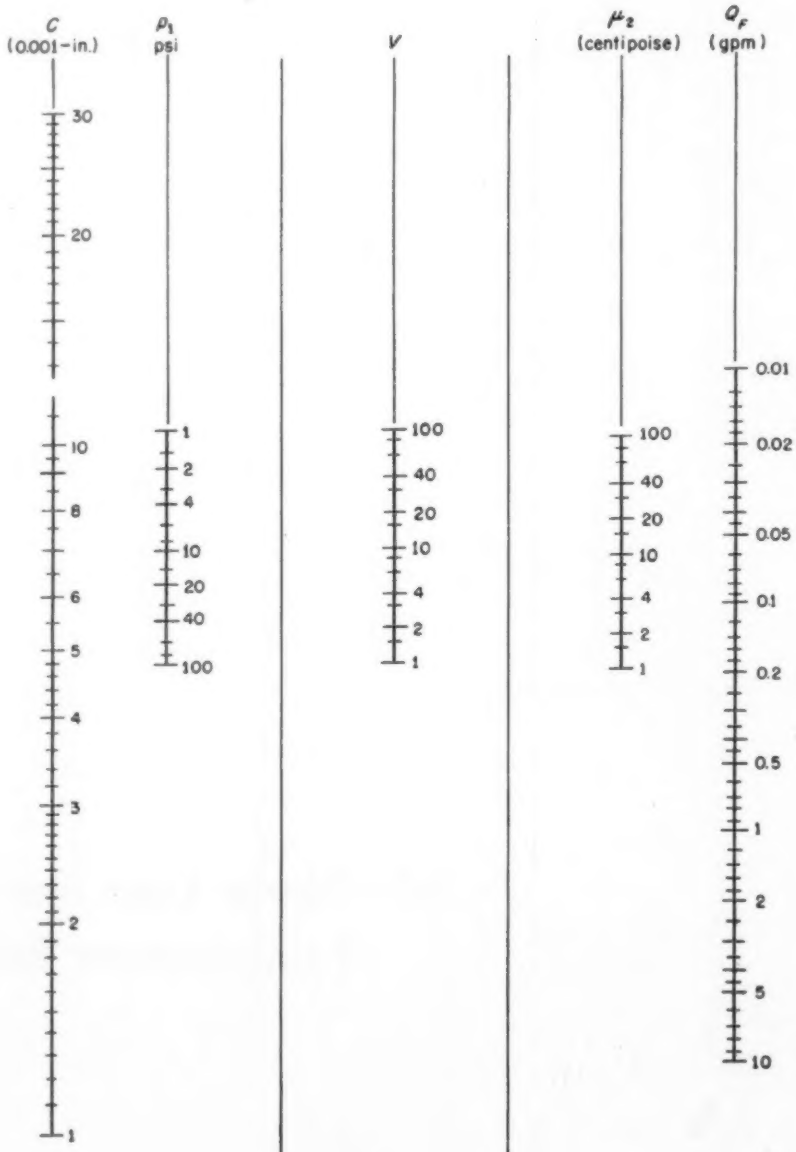


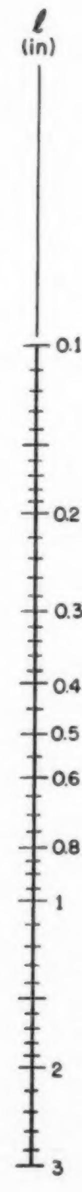


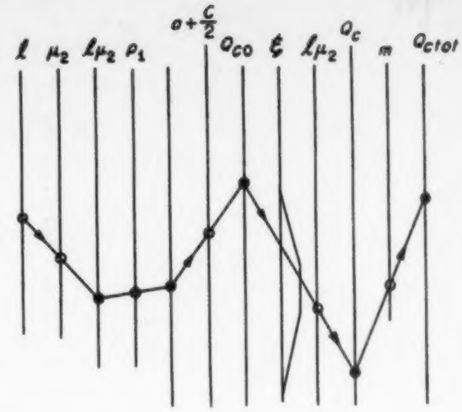


(b)—Inlet Pressure Flow

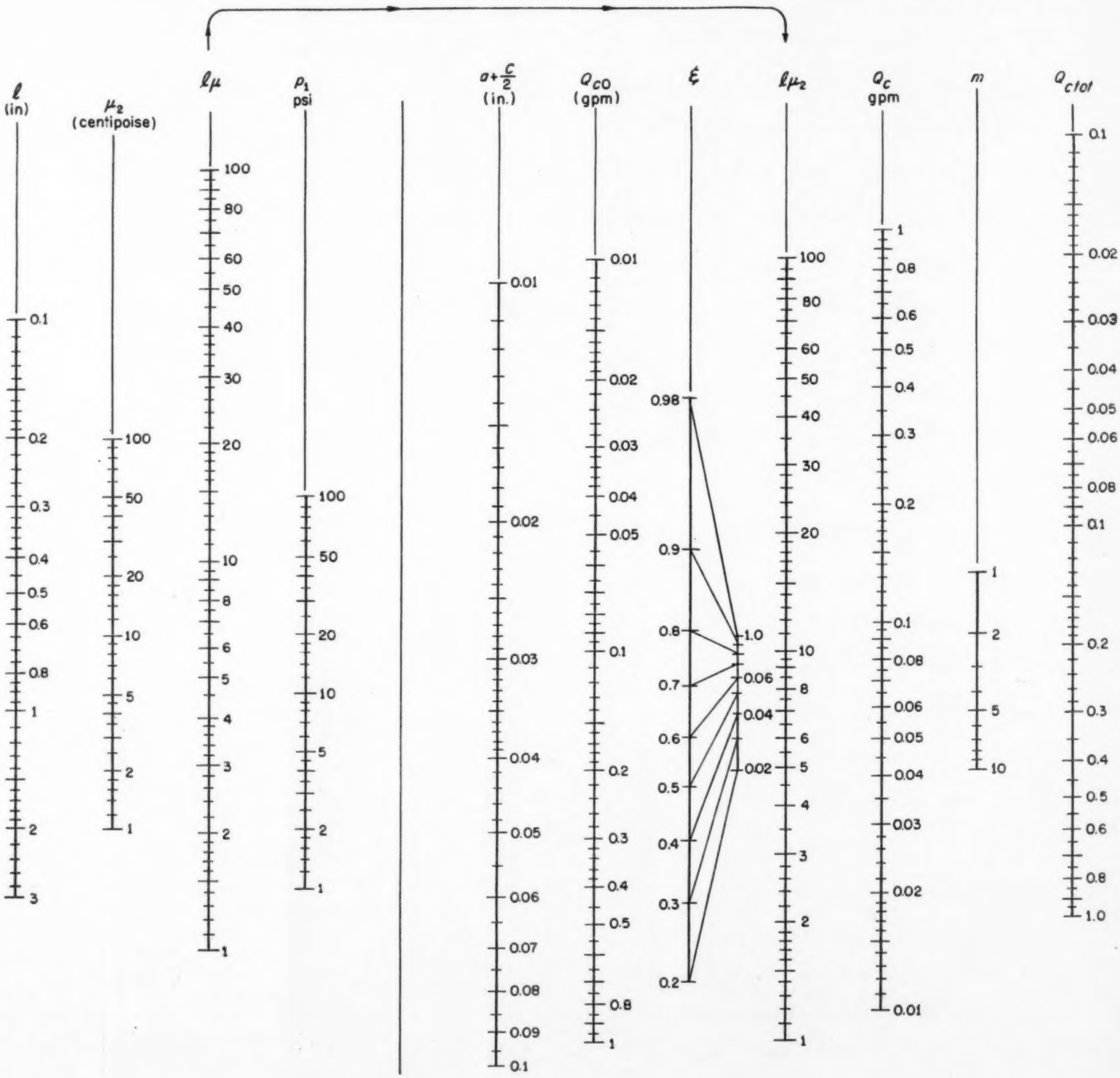








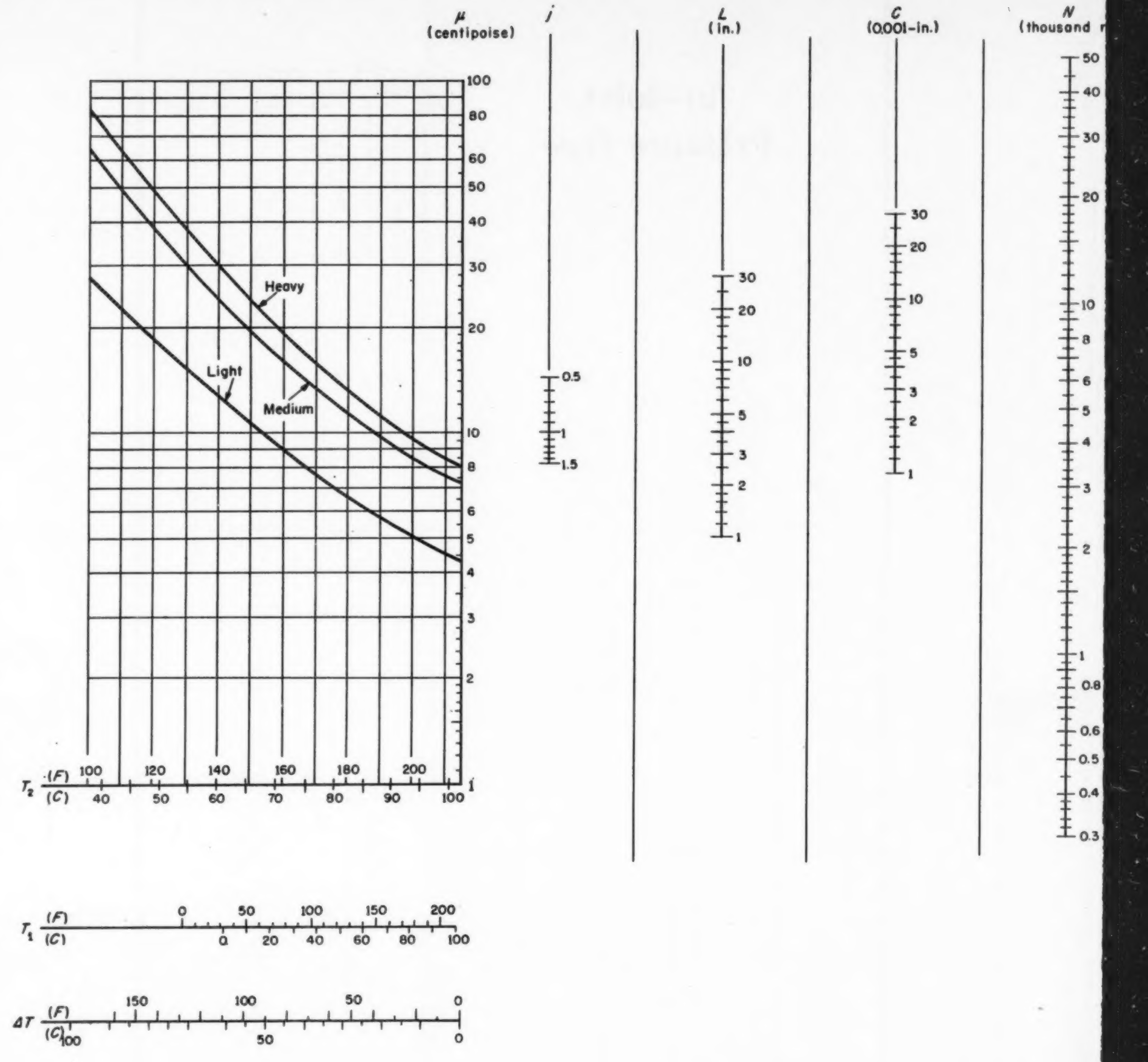
(c)—Chamfer Flow



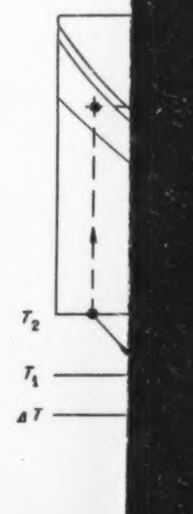
PULL OUT

Work Sheet

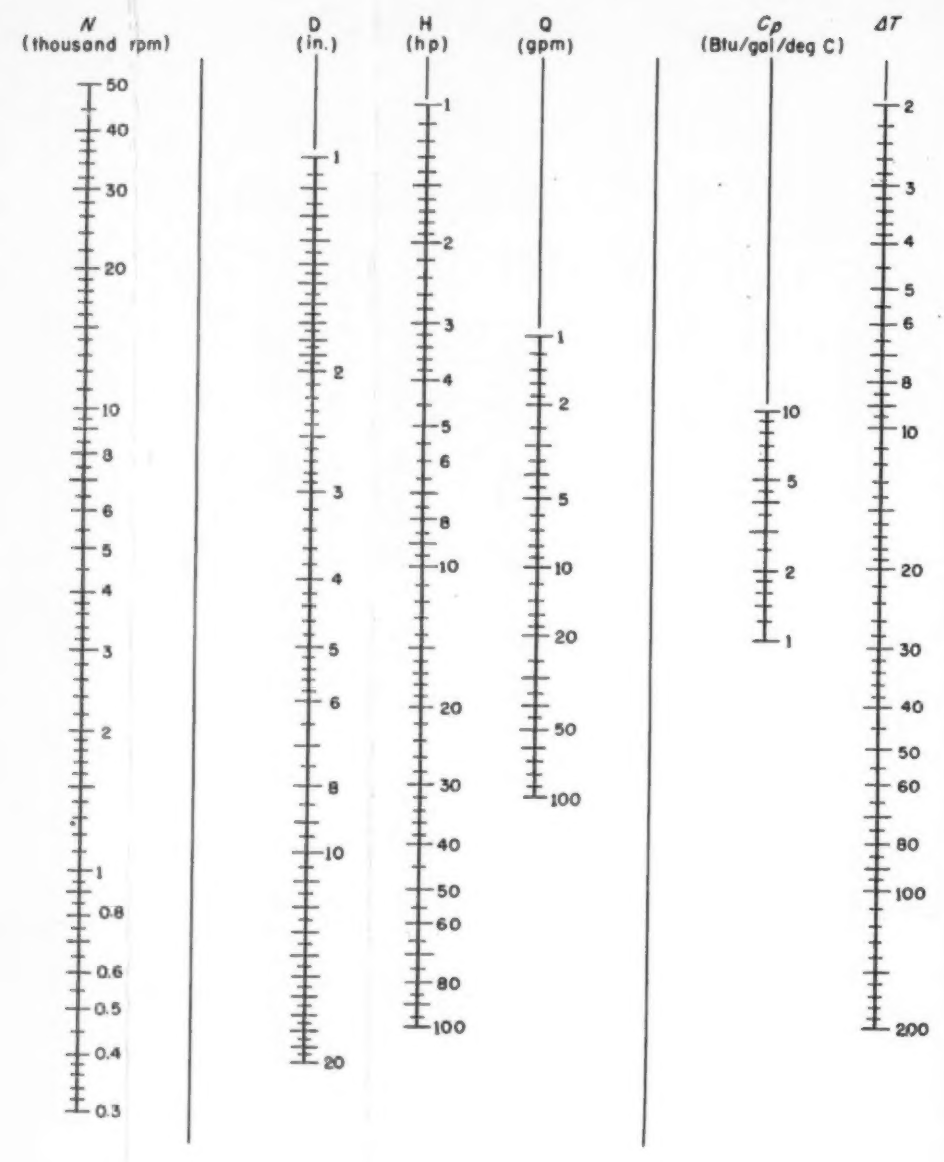
Symbol	Definition	(1)	(2)	(3)	(4)
C	= Diametral clearance, in.				
C_p	= Heat capacity of oil, Btu/gal/deg C				
D	= Journal diameter, in.				
H	= Power consumption of bearing, hp				
L	= Length of bearing, in.				
N	= Journal speed, rpm				
P	= Bearing load, psi				
Q_{c0}	= Zero speed chamfer flow, gpm				
Q_c	= Single chamfer flow, gpm				
Q_{ctot}	= Total chamfer flow, gpm (from nomograph *c*)				
Q_F	= Oil flow due to inlet pressure, gpm (from nomograph *b*) .				
Q_R	= Oil flow due to rotation, gpm (from nomograph *a*)				
Q	= Total oil flow ($Q_{ctot} + Q_F + Q_R$), gpm				
S	= Sommerfeld number				
T_1	= Inlet oil temperature, C				
T_2	= Outlet oil temperature, C				
ΔT	= Temperature rise of oil ($T_2 - T_1$), C				
a	= Chamfer dimension radial to journal, in.				
e	= Journal eccentricity, in.				
h	= Oil film thickness, in.				
j	= Dimensionless power loss ratio				
ℓ	= Axial length of chamfer, in.				
m	= Number of chamfers				
n	= Eccentricity ratio, $2e/C$				
p_1	= Inlet oil pressure, psi				
q	= Dimensionless oil flow coefficient				
v	= Width–length ratio oil film flow pattern				
μ_2	= Outlet oil viscosity, centipoises				
ξ	= Chamfer flow coefficient, Q_c/Q_{c0}				

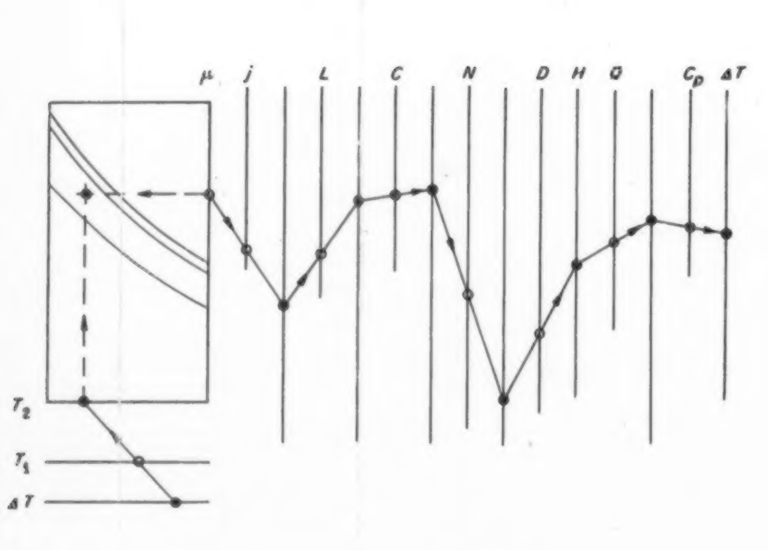


(d)—Power Loss and Temperature Rise

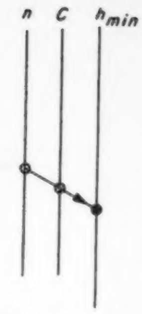


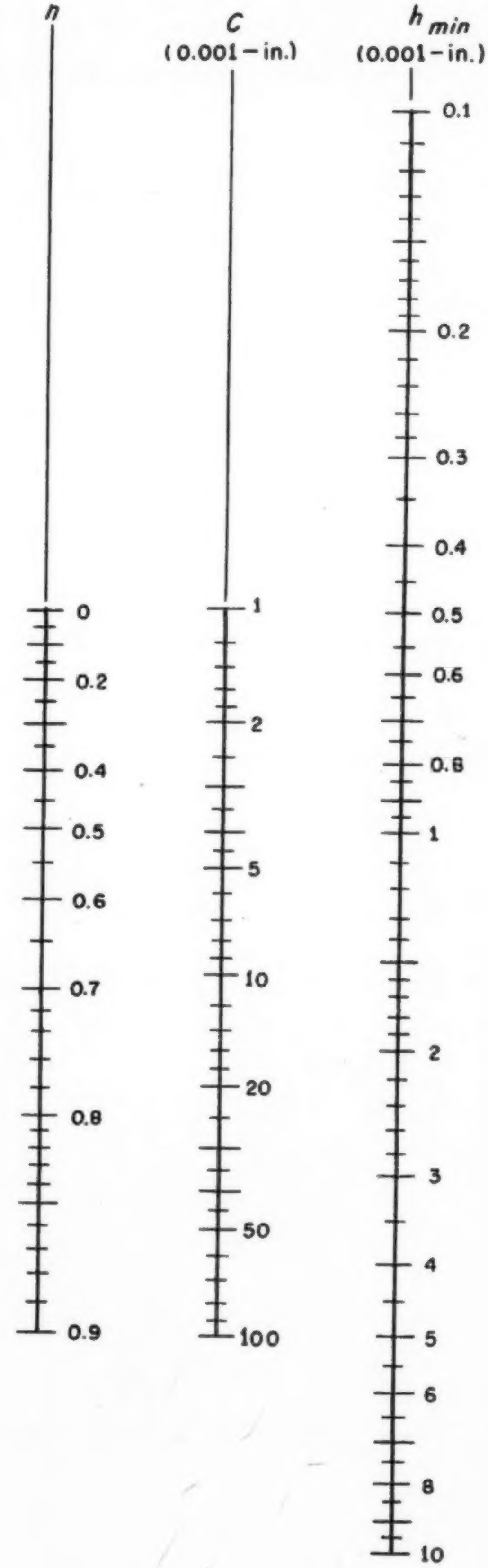
SLEEVE BEARING DESIGN





(e)—Minimum Oil Film Thickness





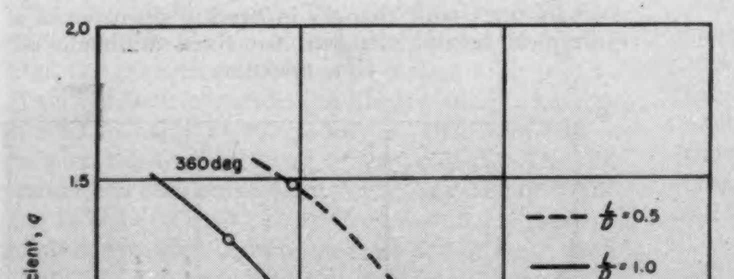
52 C for light oil, 59.5 C for medium oil, and 62 C for heavy turbine oil. Opening the chamfers of the bearing has decreased the operating temperature an average of 3.5 C or 6.3 F. Note that much of the computation is the same for both cases.

EXAMPLE 3—CHANGING INLET OIL TEMPERATURE: The influence of inlet oil temperature is very easily demonstrated at this point since the inlet oil temperature does not enter into the computation until the final step of plotting the operating line on the viscosity temperature plot. A rise in inlet oil temperature from 40 C to 60 C is thus easily demonstrated by shifting curve 1 on *Fig.* 8 to the right by 20 C, giving curve 3. This does not result in an equivalent rise in operating temperature or average outlet oil temperature for the bearing. The increase in operating temperature is smaller than the increase in inlet oil temperature as shown in TABLE 1.

EXAMPLE 4—EFFECT OF CLEARANCE: It has often been observed in bearing test work that when bearing clearances are increased the operating temperature is reduced, the oil flow is increased, but the power consumption changes very little. This experience is readily confirmed by computations made with the aid of the alignment chart. The 8 by 4 inch bearing used in the previous example running at 2700 rpm and 150 psi on medium turbine oil is influenced by changes in diametral clearance as shown in TABLE 2. What happens here is that the decreased shear rate in the larger clearance bearings is offset by the increased oil viscosity permitted by the larger oil flow.

EXAMPLE 5—EFFECT OF BEARING LENGTH: The effect of bearing length upon performance is likewise interesting. This effect is shown in TABLE 3 for the 8 inch diameter bearing with 0.0126-inch diametral clearance operating at 2700 rpm and 150 psi, using medium turbine oil. Note that the longer bearings required very little more oil flow than the short bearings and operate more efficiently, requiring fewer horsepower per inch of bearing length or per ton of load carried. However, because of the relatively constant oil flow, the longer bearings run hotter than the short bearings. Despite this fact, the minimum oil

Fig. 7—Below—Dimensionless oil flow coefficient plots for 120, 150, and 360-degree bearings



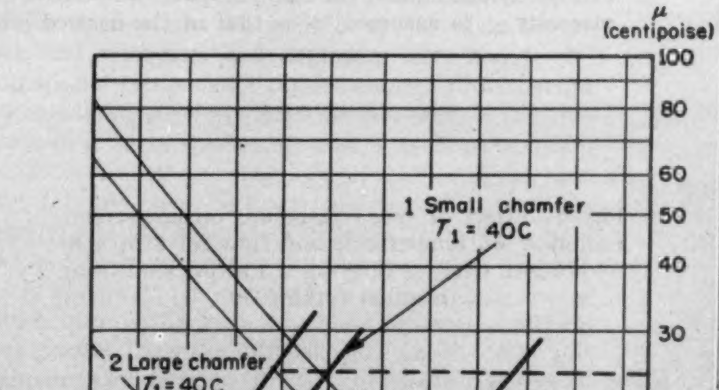


Fig. 8—Right—"Operating lines" for typical bearing as determined with nomographs of *Fig.* 6, showing effect of changing chamfer size and oil inlet temperature

film thickness is larger for the longer bearings.

EXAMPLE 6—CHANGING SPEED: The effect of speed on the performance of the same 8 by 4 by 0.0126-inch bearing is readily computed and is shown graphically in *Fig.* 9.

EXAMPLE 7—DETERMINING CLEARANCE FOR FIXED TEMPERATURE RISE: A frequently occurring design problem is to determine the proper clearance for a bearing in order to achieve a given temperature rise. Here the proper clearance for a 5 by 5 inch bearing carrying 500 psi loading and running at 4000 rpm is to be determined. The bearing has two oil grooves terminating in 0.030-inch chamfers 0.25-inch long. The oil is light turbine oil at an inlet pressure of 20 psi and an inlet temperature of 120 F. The temperature rise is to be held to 40 F.

The correct clearance is determined by first assuming a clearance and determining the temperature rise that results. When the computed temperature rise equals the desired value of 40 F, the proper clearance has been determined. In this procedure the outlet oil viscosity μ_2 is assumed to be that at the desired outlet oil temperature of 165 F, in this case 8.9 centipoises, from the viscosity-temperature chart of *Fig.* 6*d*.

A first approximation to the desired clearance may be obtained from

$$C^2 = \frac{1.03 \times 10^{-9}\,\mu_1 D^2 N}{p_1} + \left(-1 + \sqrt{1 + \frac{25.8\,p_1}{C_p(\Delta T)}\,\frac{\mu_2}{\mu_1}\,\frac{L}{D}}\right) \quad \ldots\ldots\ldots (5)$$

Substituting the appropriate values in this equation, the first approximation to the correct clearance is found to be

$$C^2 = \frac{1.03 \times 10^{-9}(18.5)(25)(4000)}{20} + \left(-1 + \sqrt{1 + \frac{25.8(20)(1.8)}{6.25(40)}\,\frac{8.9}{18.5}\,\frac{5}{5}}\right)$$

$$= 0.000064$$

$$C = 0.0080\text{-inch}$$

The approximate clearance of 8 mils is now tried out on the chart and is found to give a temperature rise of 28 F. This is smaller than the desired temperature

Fig. 9—Effect of speed variations on power consumption, oil temperature and flow for 8 by 4 by 0.0126-inch bearing carrying a 150 psi load using medium turbine oil

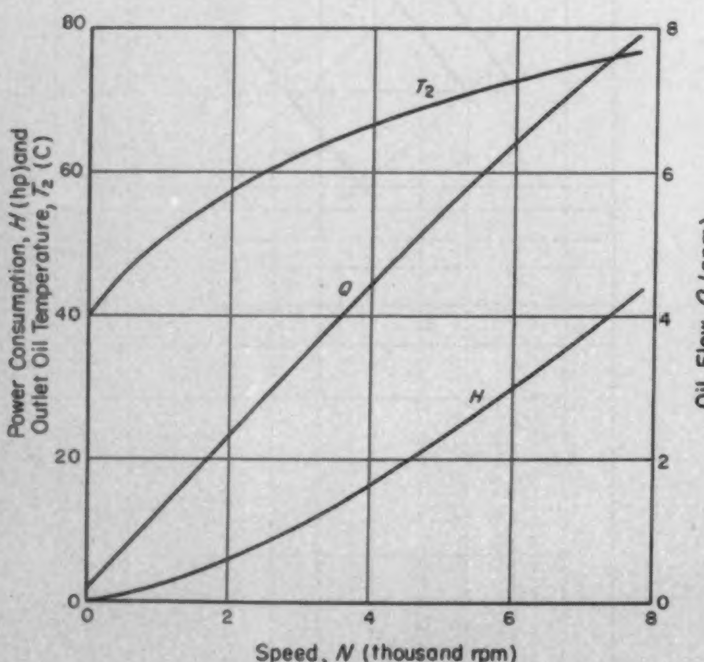
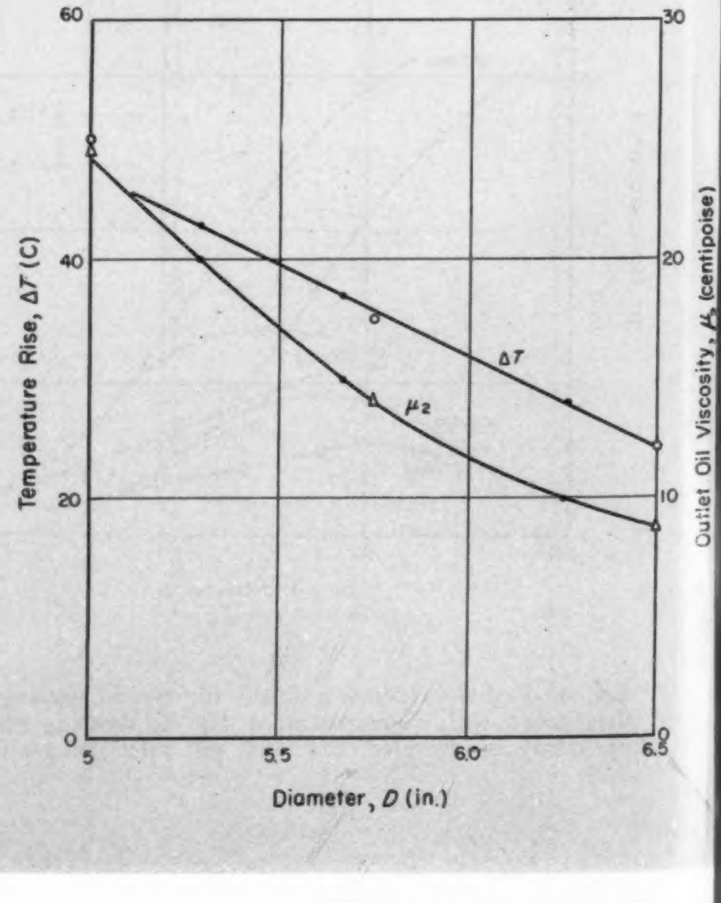


Fig. 10—Temperature rise and outlet oil viscosity variations with changes in bearing diameter in a typical bearing designed for fixed minimum oil film thickness



rise of 40 F and so a clearance of 6 mils is tried next. The result for 6 mils clearance is a temperature rise of 41.5 F which is close enough to the desired value of 40 F. The designer might, therefore, specify a clearance of 6 to 7 mils for this bearing. The oil film thickness would be 0.0010-inch minimum.

EXAMPLE 8—CHOOSING A BEARING SIZE FOR SAFE OIL FILM THICKNESS: Determination of the bearing dimensions required to achieve a certain minimum value of oil film thickness also requires a trial procedure combined with some judgment on the part of the designer. It is most readily done by assuming a value of clearance which, with the desired minimum value of oil film thickness, fixes the maximum permissible eccentricity. The further assumption of an L/D ratio for the bearing then permits the determination of the minimum value of Sommerfeld number from the q versus S chart of *Fig.* 6*a*, which also carries lines of constant eccentricity. By assuming two or three values of bearing diameter, an operating line may be charted for a constant minimum oil film thickness and constant total clearance. If total load to be carried and desired speed are fixed, the assumed diameter and L/D ratio determine the unit loading, P:

$$P = \frac{G}{D^2\left(\frac{L}{D}\right)} \qquad (6)$$

Use of the alignment chart for Sommerfeld number indicates the proper value of outlet oil viscosity for each assumed diameter. The calculation of oil flow and temperature rise is straightforward, following which the operating line may be plotted.

This type of computation is illustrated by a bearing required to carry 40,000 pounds at 1800 rpm with a minimum oil film thickness of 0.0008-inch. The calculation has been carried out for a clearance of 0.008-inch, $L/D = 1.5$, $T_1 = 40$ C, $P_1 = 15$ psi, and no chamfers (which eliminates the use of *Fig.* 6*c*). How μ_2 and ΔT vary as a function of bearing diameter is shown in *Fig.* 10. From the operating line and *Fig.* 10, the results shown in TABLE 4 are obtained for the three grades of turbine oil.

Summary: Computations of the type which have been described and illustrated herein make bearing design relatively simple, and give the designer an accurate means of comparing one design with another. The effects of changes in clearance, load, speed, base oil viscosity, inlet temperature and pressure, chamfer dimensions and oil groove shape upon the bearing performance can also be judged.

These comparisons may be made without the uncertainties which attend the experimental comparison of bearings differing only slightly. The method on the other hand may be found somewhat less accurate when an attempt is made to compare the predicted results with those found experimentally, and for sound reasons. Actual bearing performance may not agree precisely with predicted performance because of (1) inaccuracies in bearing manufacture, (2) bearing misalignment upon assembly, (3) variation in the viscosity-temperature behavior of the lubricating oil, (4) variations in oil feed temperature and pressure, shaft speed and bearing load, and (5) inability to calculate exactly the heat transfer from the bearing by conduction and radiation.

This latter point has only briefly been touched upon, the alignment chart being based on the assumption that all the heat generated in the bearing is removed by heating of the oil passing through the bearing. If the heat flow from the bearing by conduction and radiation can be estimated, its effect may be introduced into the computation on the alignment chart.

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2. Sydney J. Needs—"Effects of Side Leakage in 120 Degree Centrally Supported Journal Bearings," *ASME Transactions*, Vol. 56, 1934, Page 721.
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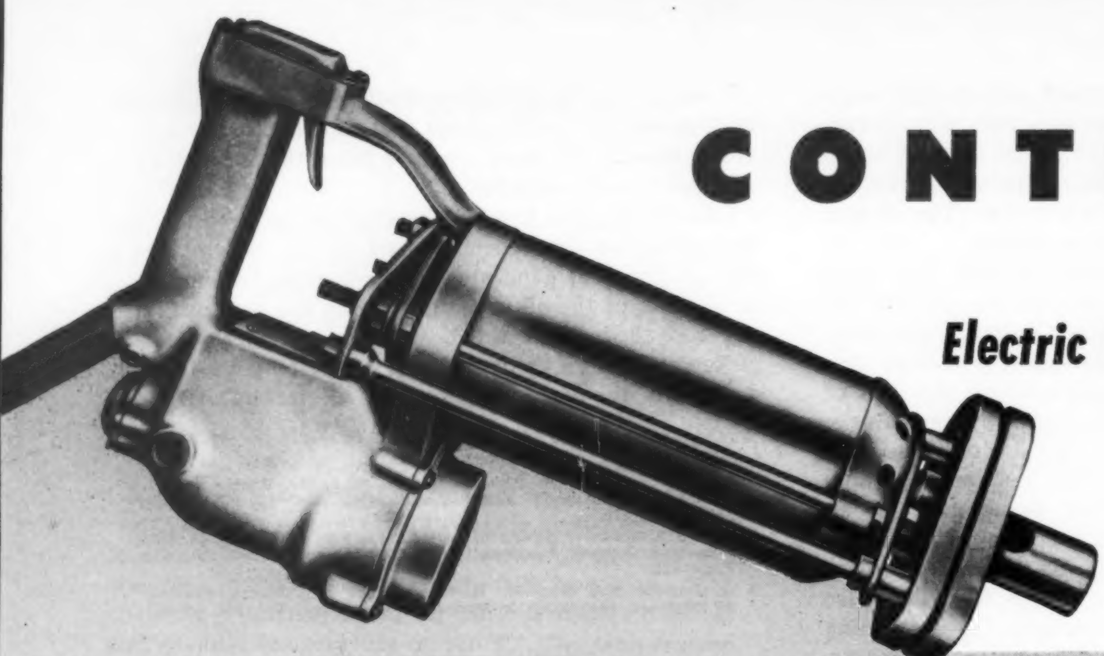
Vibration Generator

ELECTRONIC equipment is being called upon for applications where it is subjected to severe shock and vibration, and engineers have had to develop special components rugged enough to withstand such strenuous service. Electron tubes are a particular problem from the standpoint of severe vibration. In addition to being strong enough not to fail mechanically, the tube must be low in microphonics; spurious electrical signals generated by vibration of the tube must be low in relation to the desired signal that the tube is handling. In ordinary receiving tubes, microphonics may easily be a thousand or even a million times greater than the intrinsic tube noise. Tube microphonics can be a particularly difficult problem in miniaturized equipment.

An improved wide-range vibration generator is facilitating the study of low-microphonic tubes in the electron tubes laboratory of the National Bureau of Standards. The NBS tube vibrator produces accelerations up to 20 times that of gravity and is flat within 20 per cent over the unusually wide range of 100 to 100,000 cycles per second. The tube under study is fitted into a hole in the vibrator's moving element, or armature. The armature is a cylindrical block of nonmagnetic material with a "voice coil"—for the audio-frequency driving voltage—at its lower end. Two flexible metal strips hold the armature centered in an electromagnetic field structure powered by 40 watts of 120-volt direct current. Although the only model of the NBS tube vibrator thus far constructed was built to accommodate subminiature-type tubes, the design could readily be modified for microphonics studies of miniature or octal tubes.

Correction

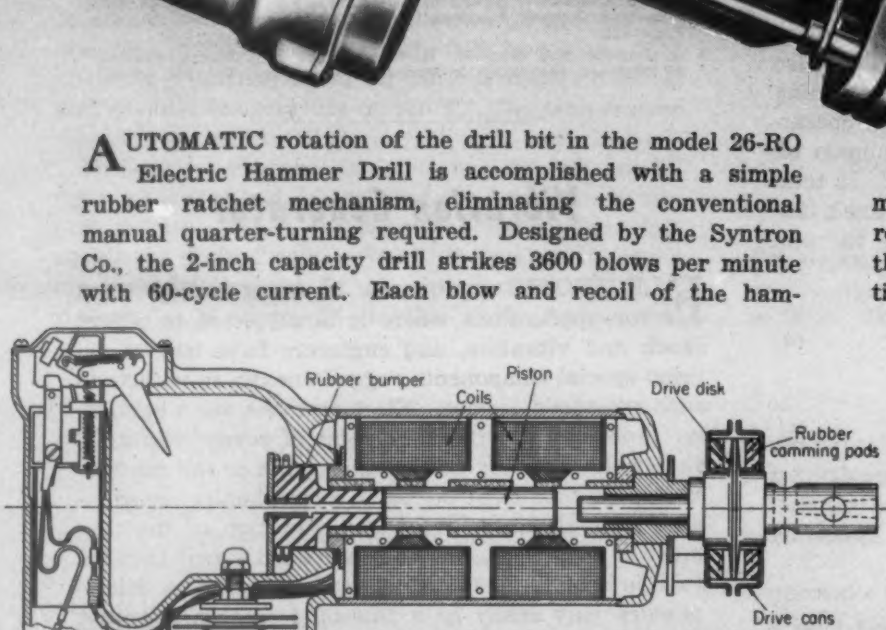
In the article, "Predicting Casting Costs," by Philip Tripoli which appeared in the June issue of MACHINE DESIGN, transposition of two charts occurred inadvertently. Charts in *Figs.* 3 and 7 should be transposed.



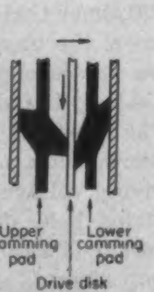
CONTEMPO

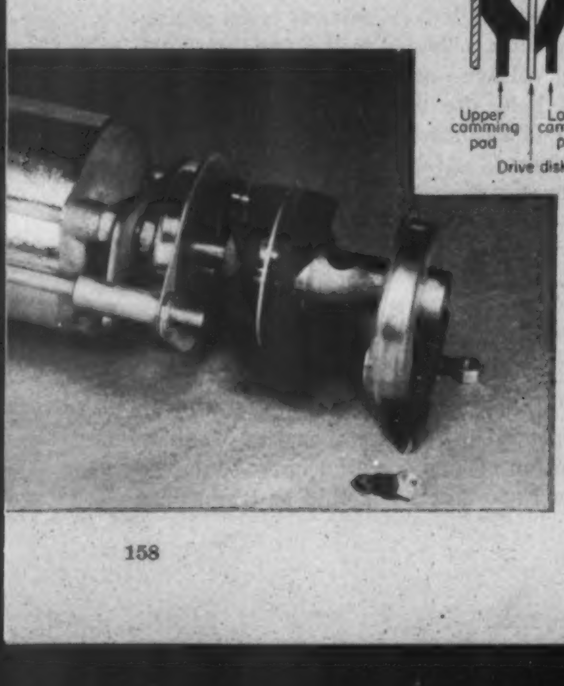
Electric Hammer Drill Bit Rotates Automatically

AUTOMATIC rotation of the drill bit in the model 26-RO Electric Hammer Drill is accomplished with a simple rubber ratchet mechanism, eliminating the conventional manual quarter-turning required. Designed by the Syntron Co., the 2-inch capacity drill strikes 3600 blows per minute with 60-cycle current. Each blow and recoil of the hammer piston causes the tool holder to rotate slightly, with considerable torque, thus producing almost continuous rotation.



Reciprocation of the steel hammer piston is accomplished by energizing two coils alternately, causing the piston to reciprocate at the frequency of the energizing current. Commercial 25, 50 or 60-cycle ac current is rectified with an integrally mounted selenium rectifier so that forward current flow is through one coil, and reverse flow through the other. Recoil of the return stroke of the piston is converted into additional hammering force by a rubber bumper, whereas "bounce" off the incompressible toolholder and tool is relatively slight. Thus, a large part of the energy of both up and down-strokes is converted into useful work.



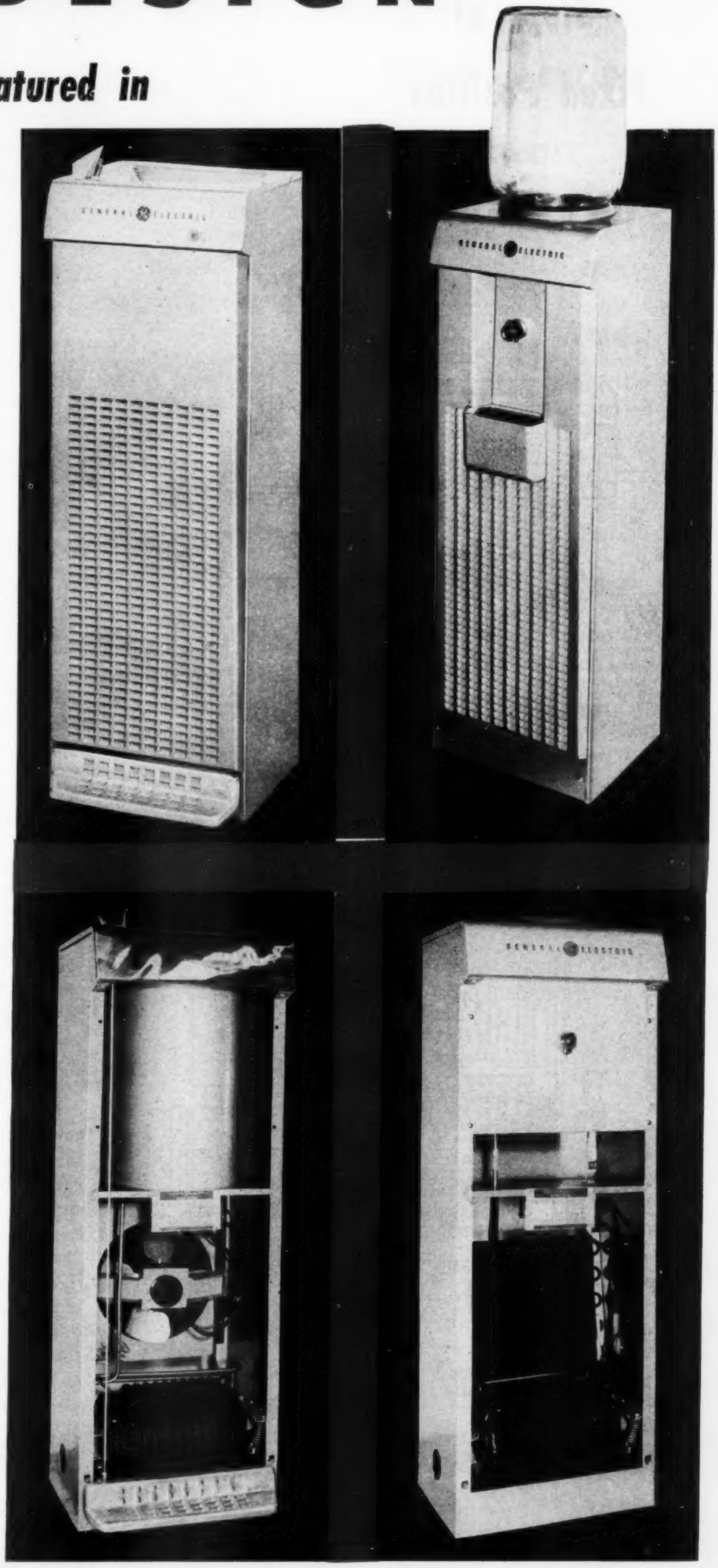


To accomplish rotation, a drive disk is shrink-fitted on the toolholder, the two pieces acting as one part. Two drive cans, housing the rubber camming pads, are clamped together to preload the camming pads, and are connected by tierods to the rubber bumper and spring at the handle end of the hammer. This entire assembly is free to move an axial distance of approximately 1/32-inch. When the piston strikes the toolholder, the drive disk compresses the legs on the lower camming pad, causing the disk to rotate slightly (see inset). As the piston leaves the toolholder, the lower legs tend to return to their original position. Since the disk is kept from rotating by the legs on the upper pad, slippage occurs between the lower legs and the drive disk. An additional rotational impulse is furnished by the "bounce" of the toolholder toward the handle end, which forces the disk against the upper camming pad to provide a second minute rotational thrust. Exactly the same sequence of actions occurs on the upstroke when the piston hits the rubber bumper, except that the drive can compress the rubber pads against the drive disk.

ORARY DESIGN

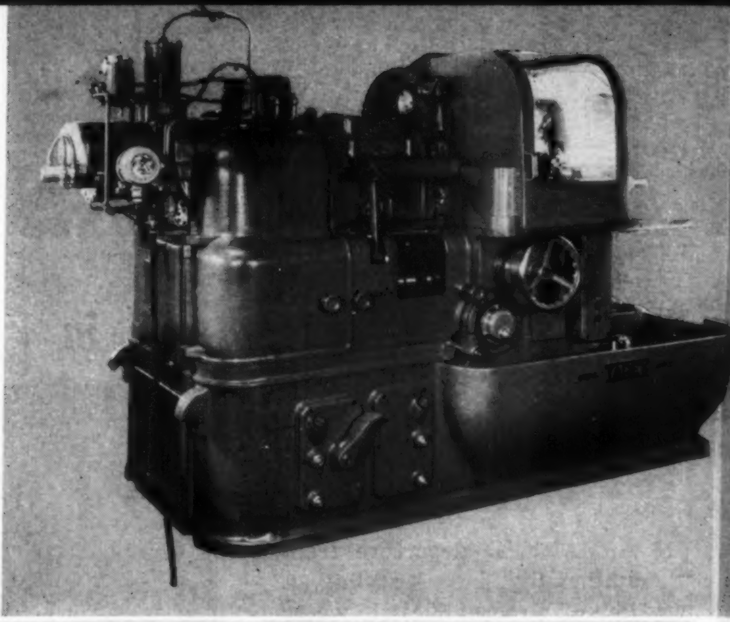
Tall, Slim Styling Featured in Water Coolers

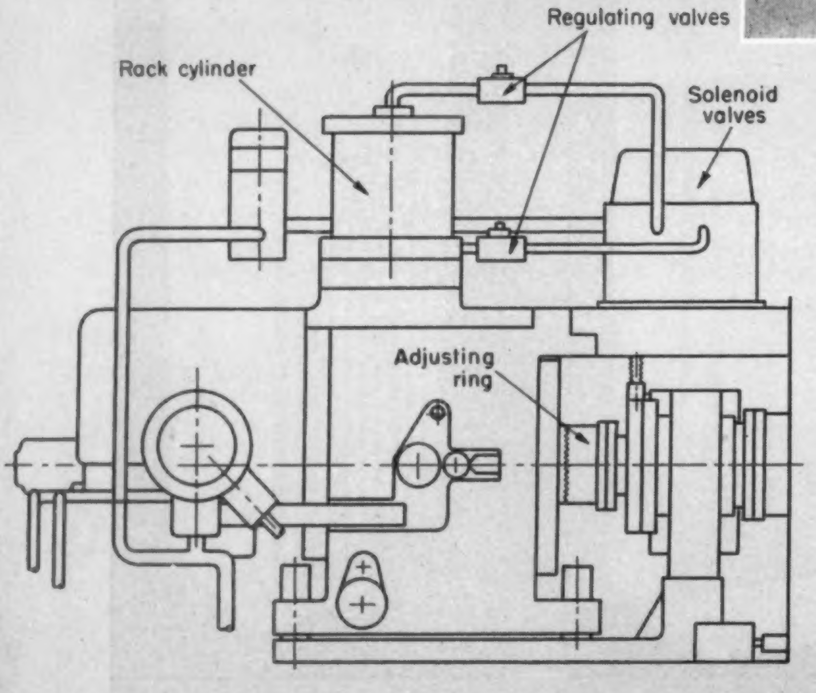
IN A MARKED departure from traditionally "boxy" styling, a new line of General Electric water coolers emphasizes tall, slim lines. Requiring 30 per cent less floor space, the recently annouced coolers are 41 inches high, but only 14½ inches wide and 13 inches deep. The pressure-type cooler features a front apron and top of electropolished stainless steel, combined with a louvered front panel that tapers in from top to bottom to a full-width pedal control. In the bottle cooler, the faucet has been recessed for safety.

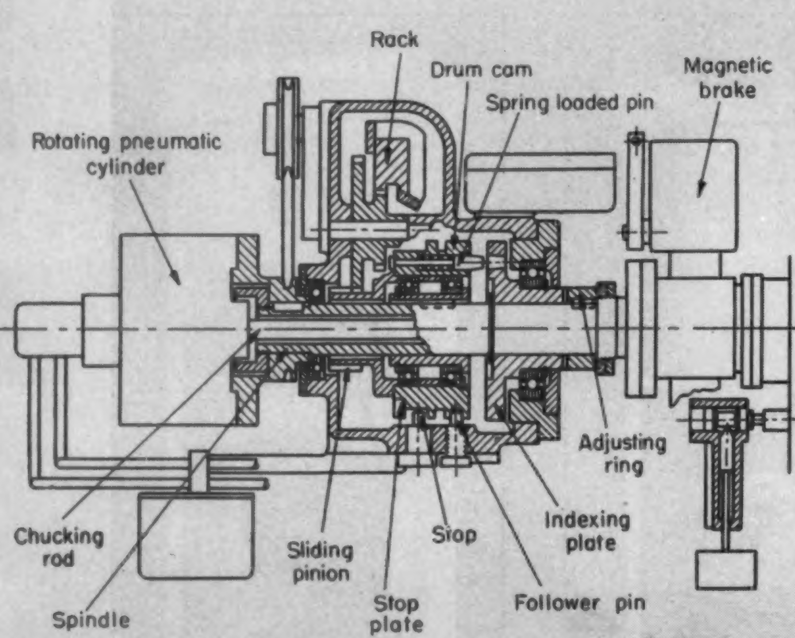


The same basic case and pan assembly is used for both models, with the physical arrangement from the middle pan down being similar. A static condenser without fan and motor is used for the bottle coolers; the pressure coolers are fan-cooled. In both types, the complete front panel snaps off for quick access to the interior controls and mechanisms. Eight water-temperature settings are available on each type, and a sealed motor-compressor is used.

Automatic Lathe Stops at Fixed Position



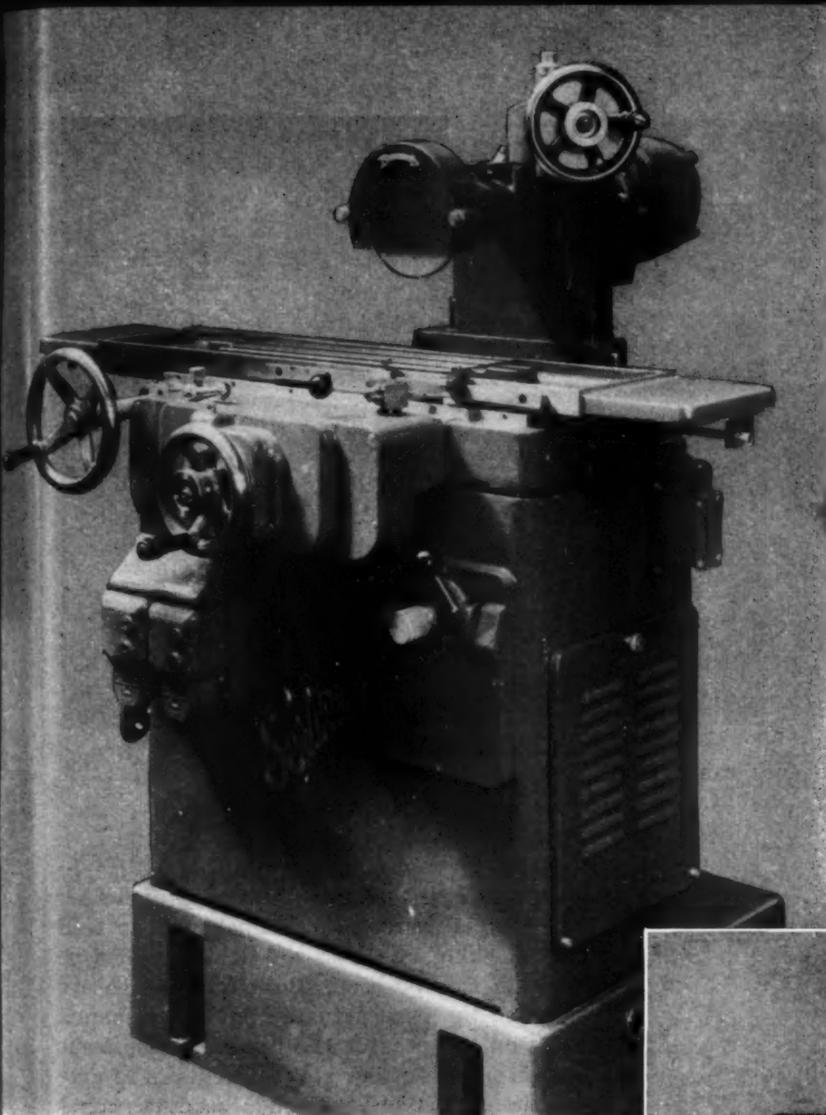




A POSITIONING device on the Tarex (Swiss) automatic lathe acts as a brake on the work spindle, and stops it at a definite angular position. The device operates at speeds lower than 10 rpm after the regular brake has been automatically uncoupled, and the final angular position is adjustable in 3-degree steps by an indexing plate.

The spindle, which rotates counterclockwise, is brought down to uncoupling speed by a magnetic brake. At this point, a rack driven by a pneumatic cylinder rotates a drum cam through a gear set. The drum cam is free to rotate and move axially along the shaft, since it is supported on the shaft by a keyed sliding sleeve and ball bearings. As it rotates, a follower pin riding in the drum-cam groove moves the cam axially. A spring-loaded pin in the cam rides over the face of an indexing plate until it enters a tapered hole, after which the indexing plate, drum cam and spindle rotate as one unit; the indexing plate is tied to the spindle through an adjusting ring. Rotation is halted at a fixed position when a stop plate on the drum cam meets a corresponding stop; a hydraulic dashpot damps out any shock.

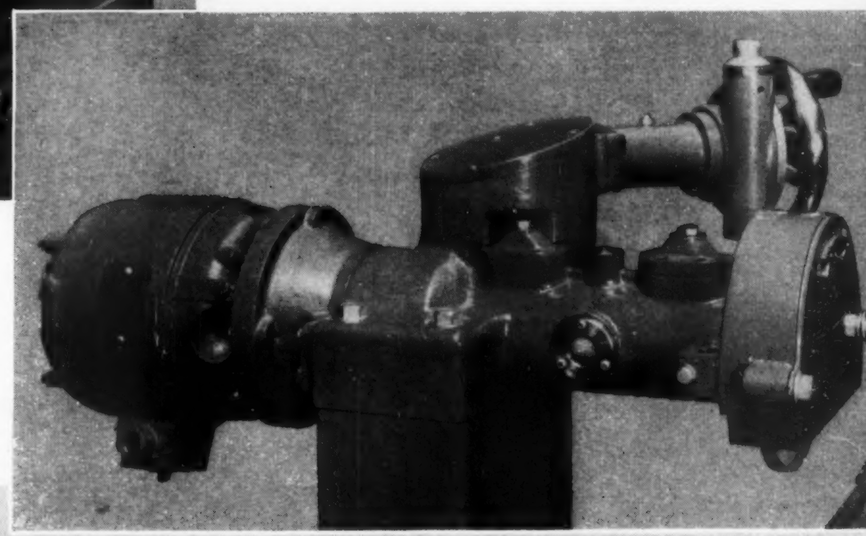
Control of the rack cylinder is by a solenoid valve, and needle valves regulate the air supply. Position of the indexing plate in relation to the spindle can be changed by backing off the adjusting ring keyed to shaft and rotating the indexing plate; axial teeth on the plate and ring tie the two units together. Chuck actuation is through a rod within the hollow spindle, operated by a pneumatic cylinder that rotates with the spindle. The spindle runs in two ball bearings, the larger of which also takes thrust loads.



CONTEMPORARY DESIGN

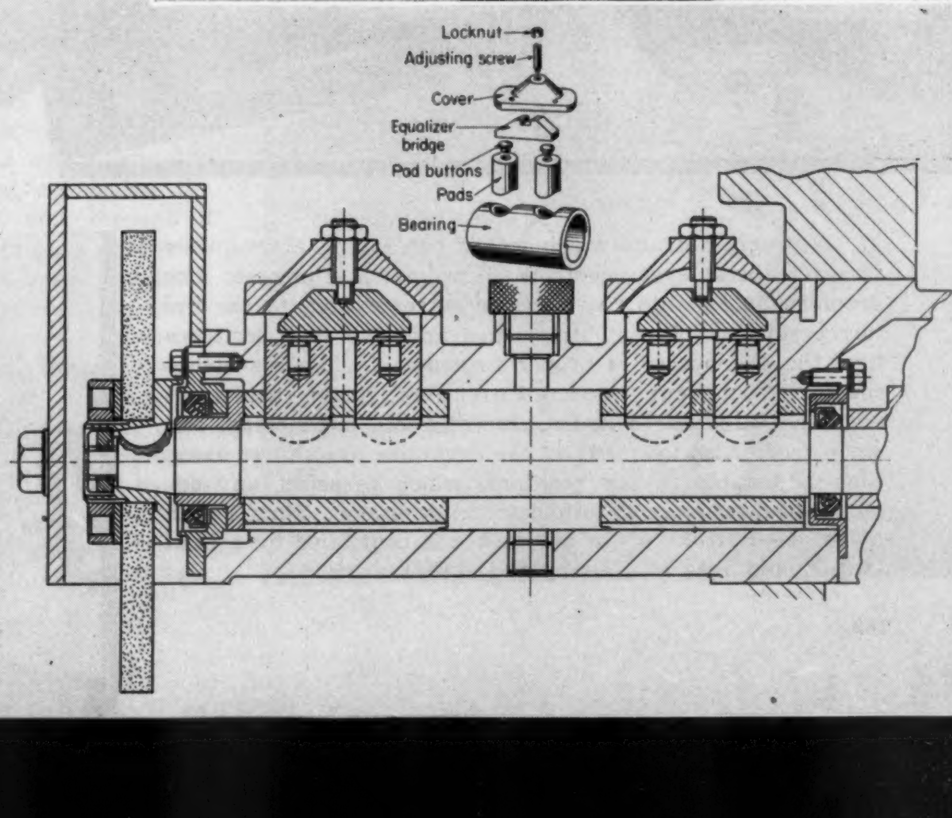
Special Bearings Increase Surface Grinder Accuracy

WIDE-SPACED 180-degree bearings, adjustable for takeup of wear, help provide accurate operation of the No. 1 Super Surface grinder of Arthur Scrivener Ltd., Birmingham, England. The 6 by 18-inch table can be operated either by hand or hydraulically, with manual or automatic reversal of speeds up to 66 fpm.

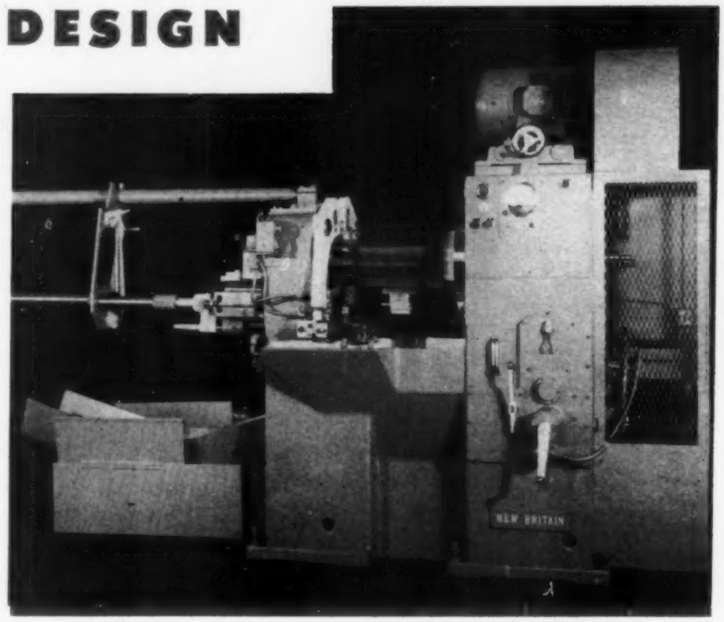


Vertical rise and fall of the wheelhead can be adjusted to 0.0001-inch, controlled by a handwheel and fine-feed attachment. The entire spindle is enclosed, and revolves at 2800 rpm in an oil bath. It is coupled directly to a 1-horsepower ac motor.

The Nitralloy spindle, $1\frac{1}{8}$ inches in diameter, runs in two long, rigid plain bearings. Adjustment is through an adjusting screw and locknut, which loads each bearing through two pads; an equalizer bridge and pad buttons distribute the "preload" equally between each pad. Thus the spindle can be kept running accurately for years despite bearing wear.

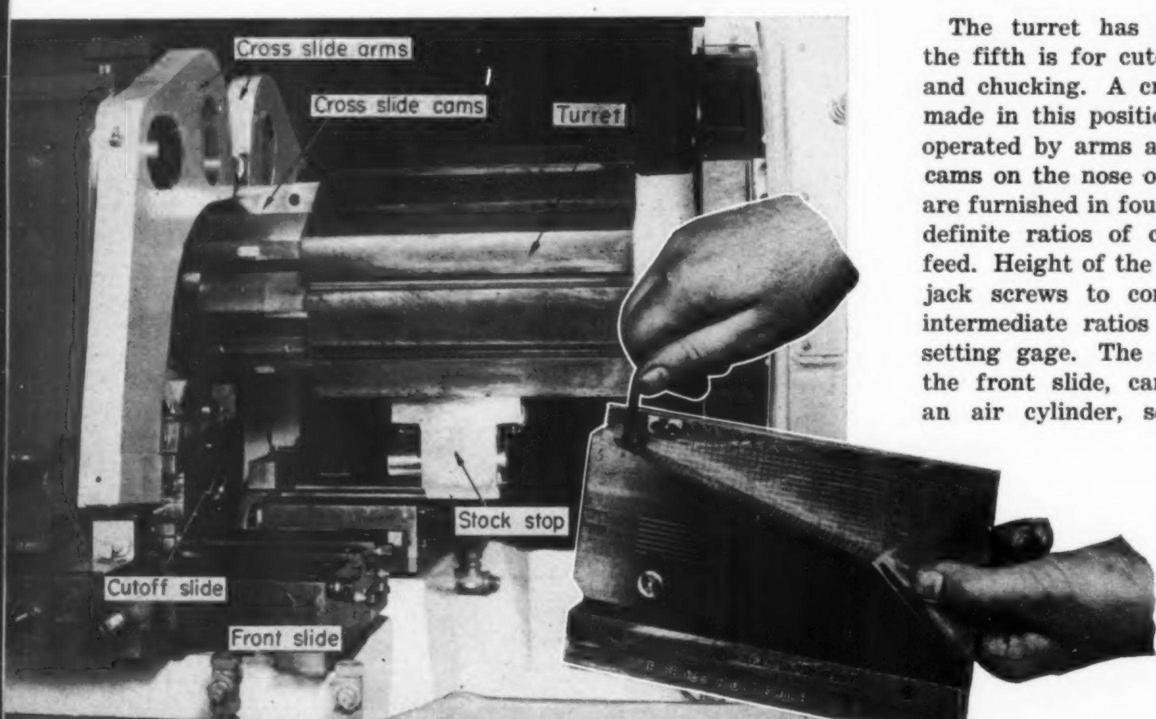


CONTEMPORARY DESIGN



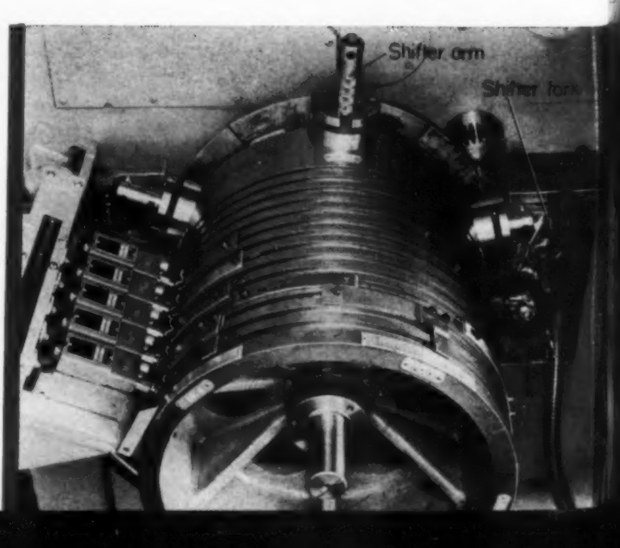
Bar Machine Designed for Simple Setups

SIMPLE operational setup—one of the design objectives in the new Model 126 single-spindle automatic bar machine of New Britain Machine Co.—has been accomplished by eliminating feed-gear changes and clutch and brake adjustments. Turret, cross-slide and cutoff feeds are correlated to the spindle speed, at definite ratios adjustable by simple cams and a speed-changing mechanism for each turret position. Spindle speed is infinitely variable over a range from 78 to 1500 rpm with an adjustable motor base and variable-pitch sheaves.



The turret has five positions, of which the fifth is for cutoff, unchucking, bar feed and chucking. A cross-slide cut can also be made in this position. Both cross-slides are operated by arms actuated by wedge-shaped cams on the nose of the turret. These cams are furnished in four standard shapes to give definite ratios of cross-slide feed to turret feed. Height of the cams may be varied with jack screws to control depth of cut, and intermediate ratios can be obtained with a setting gage. The cutoff slide, mounted on the front slide, can be moved forward by an air cylinder, so that the final cutoff operation, actuated by a fixed cam at the fifth turret position, does not interfere with use of form tools on the front slide.

A program drum, which makes one complete revolution for each turret cycle, controls all operation sequences. The drum is divided into five sections corresponding to the five turret stations. Preset strip cams and five limit switches time the movements in proper sequence; the limit switches control (from front to rear) turret feed, turret rapid traverse, the cutoff air cylinder, chucking and unchucking, and stock feed. Also mounted on the drum are five shifter arms, each adjustable to ten positions, which preselect any one of ten feed ratios in an automatic transmission. Thus, length of the feed stroke up to 6¾ inches is controlled by a limit switch, and rate of feed by the shifter arms.



Packaged Power

... for ease of maintenance and product versatility

By John L. Moody and Leo Freiberg
Mgr., Research & Development *Electrical Eng.*
Friden Calculating Machine Co. Inc.
San Leandro, Calif.

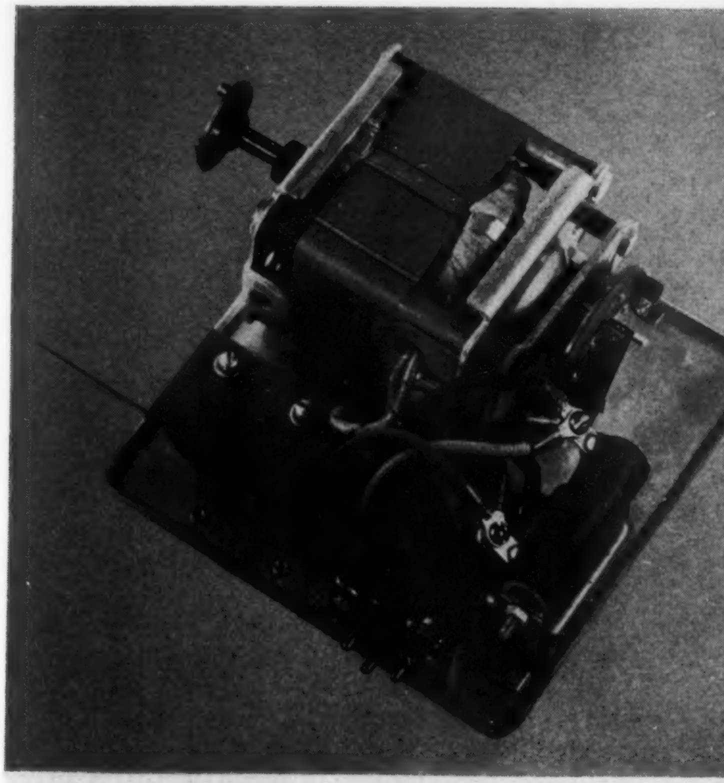
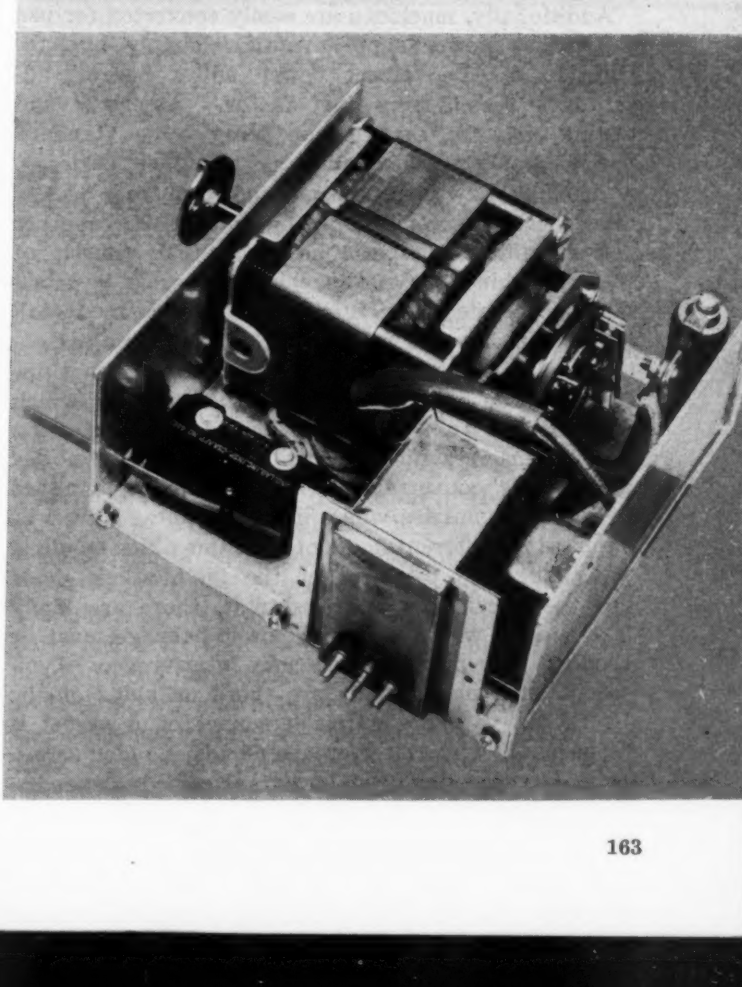


Fig. 1 — Above — Packaged-power unit components are mounted on a small plate which is easily removed or installed

Fig. 2 — Below — Interference free power package uses same basic components as standard unit and is interchangeable with it



EASE of maintenance is an essential design requirement. It may often be the deciding factor when a customer chooses between two products of comparable quality, cost and performance. Simplified servicing must be assured without sacrifice of other requirements.

Applicable in diverse forms of equipment, one way of achieving this objective is to use a packaged power unit, one that incorporates all essential power supplying components, and can be easily removed and installed. Illustrating the advantages of packaged power, this article discusses application of the concept to office equipment such as desk calculators.

The entire electrical power unit of the calculator is mounted on a single plate, Fig. 1. This equipment includes a series wound motor, governor, resistor, capacitor, switch and three-pronged connector block. Installation is accomplished by slipping the unit into the back of the machine, connecting the motor shaft to a gear reduction unit through a leather coupling and fastening with four screws.

When installed the unit is completely insulated from the rest of the calculator by insulating bushings, through which the mounting screws pass, and a sheet of 1/32-inch thick insulating material between the mounting plate and calculator base. As a further safety precaution, the unit is grounded through the center prong of the terminal block which connects through a third wire in the calculator cord to the grounding means at the wall receptacle.

In the event of a machine failure due to failure of some component of the power unit, the entire unit

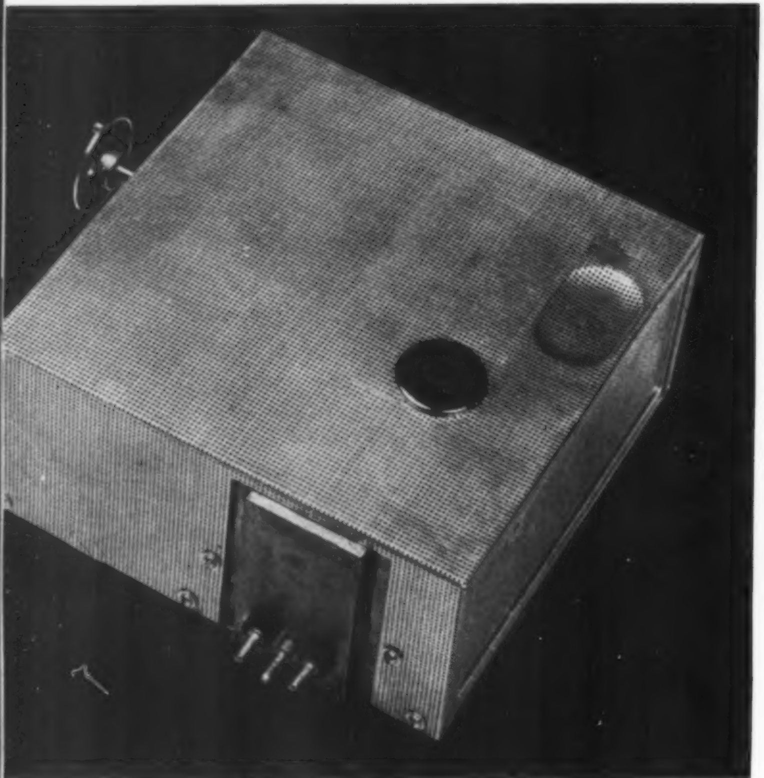


Fig. 3—Suppressed and shielded power package meets rigid specifications for low radiated noise level. Shield with round perforations eliminates noise more effectively than shield with square openings of equal area

may be removed and replaced in a matter of minutes. The faulty unit can then be taken to the repair shop for reconditioning under more favorable circumstances.

Additionally, machines are easily converted for use on various power supply voltages simply by substitution of the correct power unit. Several advantages result from this factor. Assembly and testing may be carried out using some standard power supply voltage, such as 115 v 60 cycle ac. A machine may then be converted to operate on any other desired power supply by substituting another power unit. Distributor inventory can consist of fewer machines, as one machine can easily be modified to operate on a variety of voltages by installation of the proper power unit. If, for some reason, the customer should wish to convert his machine for operation on a voltage other than that for which the machine was originally supplied, the conversion can be made simply by installing another power unit. This convertability is especially appreciated in foreign countries where different voltages may be supplied on different floors of the same office building.

Although 90 per cent of the installations made use the previously described unit, there are many instances in which a special power package must be used, to prevent radio frequency interference. Typical of such installations are those on ships of the U. S. Navy. The entire personnel of a vessel is jeopardized when interference clouds a radar screen or makes reception of an important radio message impossible. Radio frequency interference cannot, therefore, be tolerated.

The interference problem was solved by a suppressed and shielded power unit Figs. 2 and 3, which meets rigid requirements for both radiated and conducted noise level. This unit is interchangeable with the standard unit.

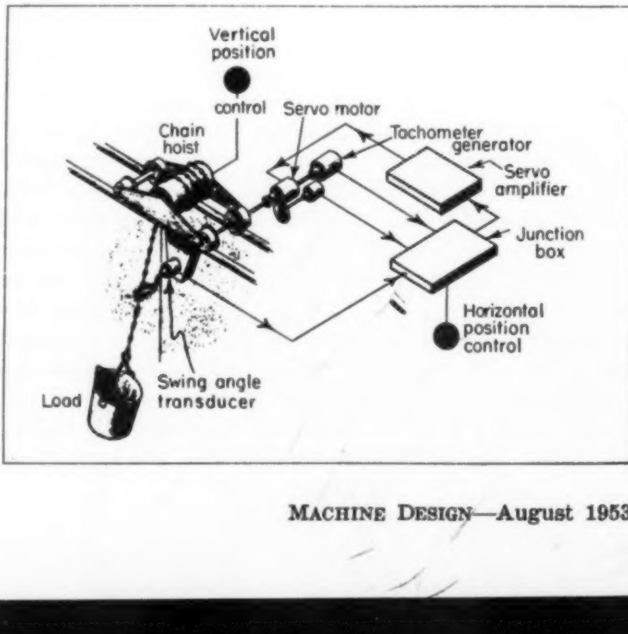
"Hydraulic Control Systems" Reprints Available

The five-part series, "Hydraulic Control Systems," by R. Hadekel is being reprinted owing to numerous requests for these articles. Compiled under one cover, the series is being made available at one dollar per copy with discounts on quantity orders. Address Readers' Service Dept., MACHINE DESIGN, Penton Publishing Co., Cleveland 13, Ohio.

Crane Swing Eliminator

TO ELIMINATE dangerous crane swinging when thermally or radioactively "hot" materials are being handled, the Argonne National Laboratories have developed an antiswing device which may be added to a conventional crane and hoist at relatively low cost. The mechanism automatically moves the crane carriage in a direction opposing that of the swing to bring the load to complete rest in only one swing at the exact position desired—load "jockeying" is unnecessary.

A transducer and servomechanism operate the crane drive to produce the desired result. Angle of swing is converted to a proportional electric current which feeds the servomechanism. A servomotor then moves the crane carriage the exact amount required to stop the load swing. The mechanism has been successfully operated in model form at the Argonne Laboratories.



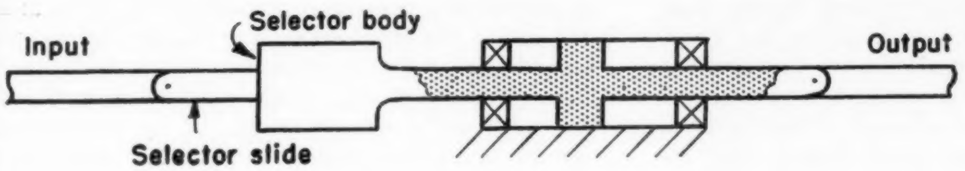


Fig. 127—Simple mechanically linked servo arrangement of a selector body mounted on the piston rod

HYDRAULIC CONTROL SYSTEMS

Basic design of machine hydraulic systems is put on a methodical and consistent foundation in this series of articles to aid development of circuits for maximum simplicity and effectiveness

Part 5–Servo Systems

By R. Hadekel *Consulting Engineer, London, England*

A SERVOMECHANISM may be defined as one kind of system intended to cause a controlled variable, the output, to follow closely a controlling variable, the input, by measuring the difference between the two, the error, and causing this difference to alter the output in such a sense that the difference will tend toward zero. Systems of this type may be divided into *regulators,* in which the input is normally fixed, and *servomechanisms,* in which the input is normally subject to more or less arbitrary variation. Many of the automatic valves encountered in this series are regulators by this definition, including relief valves, reducing valves, and metering valves. Thus a relief valve compares the pressure load on the spool or output with the spring load or input, and according to their difference opens the passage until output and input are equal.

There is a general theory of servomechanisms and regulators, which is applicable whatever the type of component from which the system is built up, be it mechanical, electrical, hydraulic, or pneumatic, and whatever the type of variable under control—position, speed, temperature, etc. This theory is concerned chiefly with the means of achieving the desired degrees of stability and accuracy, and is too complex to be even sketched out here. (See References 5, 6 and 7, Part 1.)

By far the most important type of servo system, and the only one which will be considered here, is the positional servo—that in which the controlling and controlled variables are displacements of mechanical members. The simplest and often the most convenient way of measuring the error is by means of a mechanical linkage. To avoid a discussion in abstract terms, the problem will be illustrated with actual solutions for the case of hydraulic servos, consisting of a selector and cylinder, pressure fluid being continuously available. If the selector body is mounted on the cylinder piston rod and the selector lever or slide is connected to the input linkage, as shown in *Fig.* 127, the desired objective is achieved since the actual degree and sense of opening of the selector depend on the error, or the difference between motions of the input link and cylinder and the latter will be actuated until

the error is brought to zero.

This linkage may be the simplest possible, but in practice is not always the most convenient because oil must be brought to the selector through flexible lines which must be capable of accommodating the whole stroke of the cylinder. Two alternative linkages are shown in *Fig.* 128, and these avoid the problem of flexible lines by allowing the selector to remain fixed.

In the linkages shown, the input load is that required to operate the valve, which may be limited to frictional forces. In some systems it may be desirable to have "feel"—to make the input load equal to a fixed fraction of the output load—and this can be done by using a valve for which the operating load is proportional to the pressure in the cylinder lines. An example of such a valve is shown in *Fig.* 129. Exactly the same result can be achieved by purely mechanical feedback of load—by a modification of the linkage of *Fig.* 128*a* as shown in *Fig.* 130. If the input motion is one of rotation over a large range of angles, as for instance in vehicle steering gear, a convenient linkage is formed by an epicyclic gear train, as shown in *Fig.* 131.

Usually the amount of error is limited by the maximum travel on the valve or equivalent controlling element. The input speed cannot then exceed the maximum possible output speed, which is limited by the characteristics of the system. In some cases such as aircraft flap controls it is desirable to allow unlimited error; the input lever is set to the desired position and the output is allowed to follow up in its own type. This action can be easily achieved by allowing the valve sufficient overtravel, or by interposing a cam or equivalent mechanism to allow large errors without correspondingly large valve travels.

Problems of Hydraulic Servos: The selector and cylinder combination discussed in the preceding section is but one of the many possible components which could be used for the purpose. The linkage of *Fig.* 127 is probably limited in practice to a cylinder and selector system or to close equivalents thereof, but the other linkages shown apply in principle to any system. In such systems the selector is replaced by a general controlling element and the cylinder by a general power element. In hydraulic servo practice, the power element—motor or output element—can hardly be anything but a cylinder or a hydraulic motor plus reduction gear. The only two types of hydraulic position servo of practical interest at the time

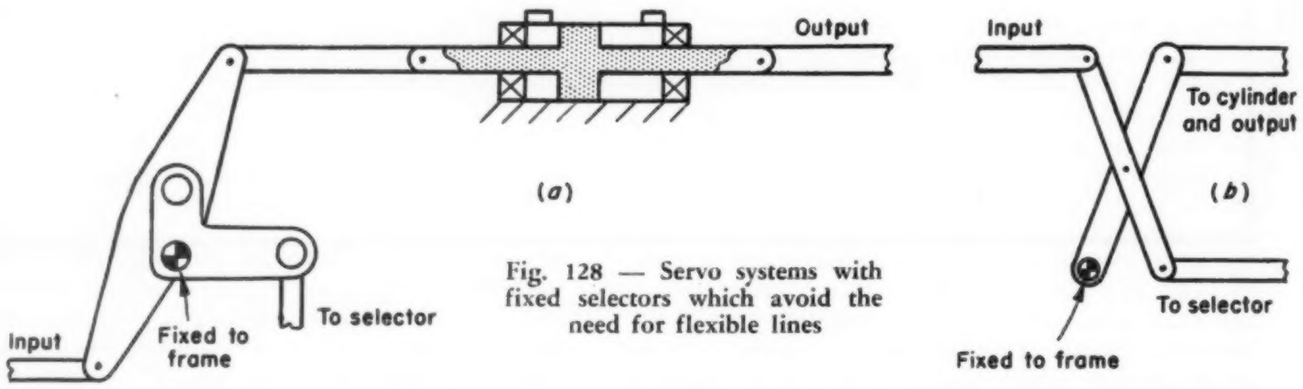


Fig. 128 — Servo systems with fixed selectors which avoid the need for flexible lines

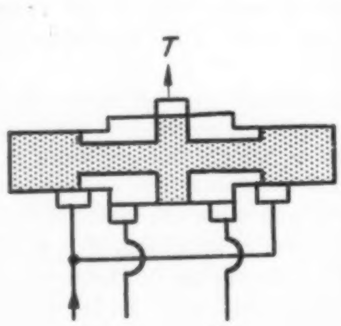


Fig. 129 — Valve design with input load set at a fixed fraction of the output load to provide "feel"

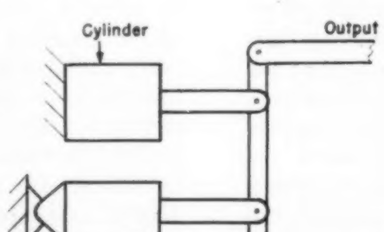


Fig. 130—System modification of that in *Fig.* 128*a* to afford "feel" by means of mechanical feedback

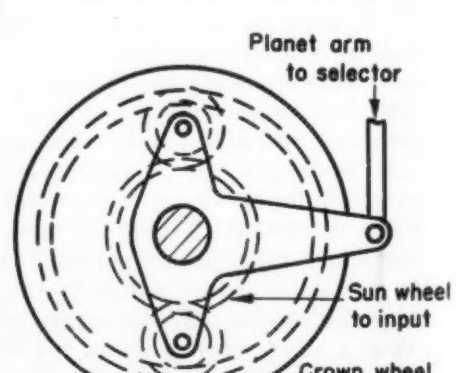


Fig. 131 — Epicyclic gear train linkage to provide feedback for rotary control

of writing are those based on control by a selector or resistance, and those based on control by a variable-delivery pump.

If the controlling selector is a slide valve with definite overlap in neutral or a seating valve with definite clearance between lever and valve push rods, there will be a small zone of error to which the control is insensitive, and high accuracy cannot be obtained. To get maximum accuracy, it is necessary to use a slide valve with negative overlap or underlap as it is sometimes called, such as the type shown in *Fig.* 54*b*, Part 1. The same effect is obtained with a seating selector with negative push rod clearance, i.e., a selector in which all valves are slightly lifted from their seats in the central position. In some low power systems such as copying machine tools, it may be more convenient to use the circuit of *Fig.* 54*a*, the controlling element being a throttle valve instead of a selector. With this system zero steady-state error can be achieved only if the cylinder load (including friction) is zero. The same applies to negative overlap selectors within the range over which negative overlap exists. In practice, however, systems of this type can be made extremely accurate, since only very tiny errors are required to develop very large cylinder loads, i.e., the "stiffness" of the system is high. With position overlap or push rod clearance, the stiffness is zero within the overlap range and infinite beyond it since, however small the selector opening, the full cylinder load will be developed once the system has stopped moving, and there is no pressure loss due to throttling.

Part of the problem of servos consists in regulating the speed of the output to equal that of the input, which may have any value down to zero. In the systems so far examined, this regulation is achieved by throttling the flow to the cylinder with consequent dissipation of power. Unless the peak power is low, this dissipation can usually not be tolerated and the question of unloading arises. Many of the unloading methods discussed in earlier articles are inapplicable to servo systems for obvious reasons.

Use of open-center selectors in the normal manner is subject to the following conditions:

1. The selector must be spring centered and the centering spring powerful enough to overcome friction in the input linkage.
2. The system must be irreversible.
3. The input member must be released to allow the system to be unoaded.

The third condition is impracticable in many, perhaps most, servo controls.

If the selector is not spring centered and the input linkage is not released, the control will stabilize at a position in which the by-pass path is throttled to the exact extent necessary to produce a pressure corresponding to the cylinder load, which may be acceptable if the operating conditions are such that the load is fairly small when the cylinder is stationary. The travel required to close the open-center path, however, may still be excessive and involve large steady-state errors. If the steady-state load is appreciable, though residual pressure will be maintained in any event and, hence, the unloading passage can be made quite small. The resulting valve would be intermediate between the open-center selector of *Fig.* 132 and the negative overlap selector of *Fig.* 54*b*. The passages in neutral would be large enough to accommodate the full pump flow at substantially less than the peak pressure, but would not be so large as to reduce pressure to nearly zero in the neutral position and the resulting error or stiffness may then well be quite acceptable.

Another possible partial solution would lie in the use of an accumulator and cut-out system. Both here and with the open-center system, the resulting power dissipation may still be excessive in many cases. The problem of power dissipation in servo systems with speed control by throttling is still unsolved — hence the attraction of variable-delivery pump systems. In such systems power dissipation is limited to losses in the pump which may, however, be far from inappreciable if large steady-state loads have to be resisted. The usual variable-delivery pump servo utilizes the basic system of *Fig.* 133, control being from the error link. A basic characteristic of this system is a "dead zone" effect if the cylinder load is zero or includes friction. In some systems* this effect is eliminated by super-imposing a high-frequency dither on the delivery control member.

The foregoing sections have barely skimmed the surface of the subject of hydraulic servos, which is regarded as too specialized to warrant more extensive treatment in this series. Various more advanced aspects will be treated in later articles to be devoted to this subject.

Port Valve Friction and Hydrodynamic Loads: A subject of particular importance in connection with servos is that of valve operating loads. In theory, if a slide valve is balanced in the sense of achieving

* G. C. Newton Jr.—"Hydraulic Variable Speed Transmissions as Servo Motors", *Journal of the Franklin Institute*, Vol. 243, No. 6, 1947.

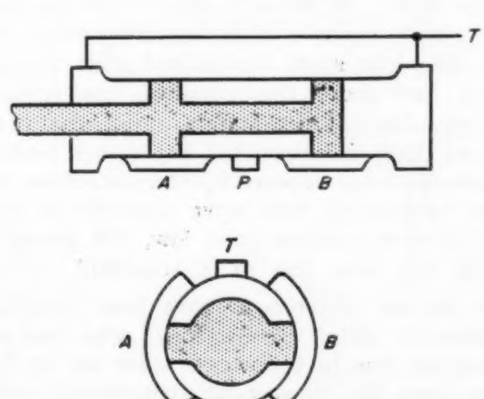


Fig. 132—Open-center selector with large negative overlap as compared to that of the selector of *Fig.* 54*b*

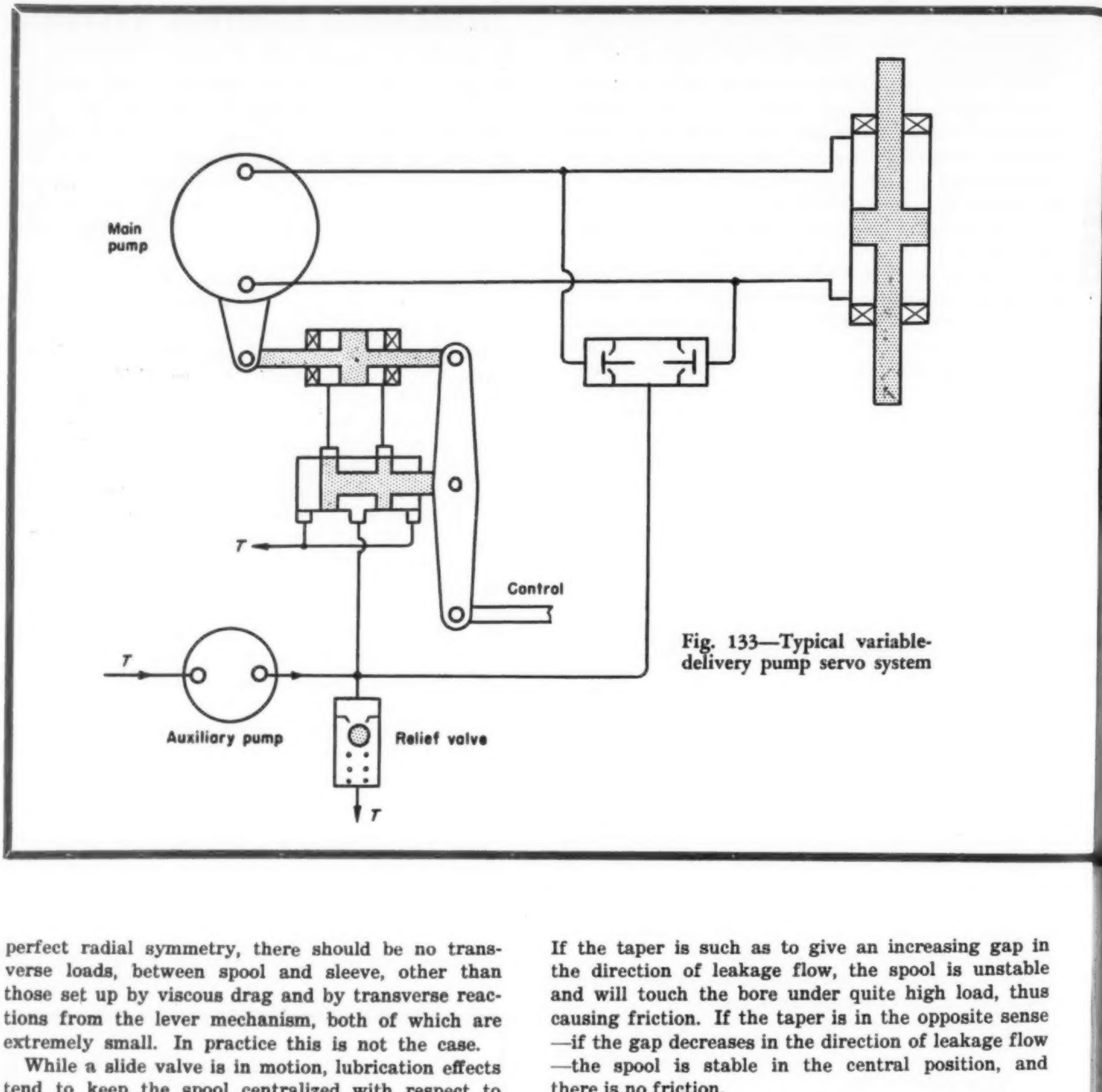


Fig. 133—Typical variable-delivery pump servo system

perfect radial symmetry, there should be no transverse loads, between spool and sleeve, other than those set up by viscous drag and by transverse reactions from the lever mechanism, both of which are extremely small. In practice this is not the case.

While a slide valve is in motion, lubrication effects tend to keep the spool centralized with respect to clearance, and under this condition the friction is indeed very low. When motion stops, the spool is free to set itself eccentric, but any forces tending to have that effect are opposed by viscous forces. Hence for some appreciable time after cessation of motion, starting friction remains quite low, but grows after the valve has been left at a standstill.

Valve friction phenomena have been investigated in considerable detail by Sweeney† who has shown that they are due to transverse loads set up by departures from the theoretical symmetrical pressure distribution in the leakage path, caused by minute taper or other imperfections of the spool and/or sleeve surfaces. It is noteworthy that the spool has a self-aligning action in that it sets its axis parallel to that of the sleeve, whether or not it is eccentric. If the taper is such as to give an increasing gap in the direction of leakage flow, the spool is unstable and will touch the bore under quite high load, thus causing friction. If the taper is in the opposite sense—if the gap decreases in the direction of leakage flow—the spool is stable in the central position, and there is no friction.

In accordance with this theory, it is found that friction loads can be substantially reduced by interrupting the sealing land by one or more circumferential grooves, as shown in *Fig.* 134, which have the effect of tending to restore symmetry of pressure distribution. It has also been suggested that additional and possibly much worse friction effects may be associated with dirt in the oil, but further evidence is needed before this can be accepted.

In servo applications—where it is particularly essential to keep valve operating forces to a minimum—friction is often eliminated by giving the sleeve a movement of rotation relative to the spool, or by superimposing a high frequency dither on the motion of the latter. As implied previously, friction can also be eliminated by giving a suitable taper to some of the valve lands. Yet another method is shown in *Fig.* 135. Pressure is admitted through paths of high

† D. C. Sweeney—"Preliminary Investigation of Hydraulic Lock", *Engineering*, 26-10 and 9-11, 1951.

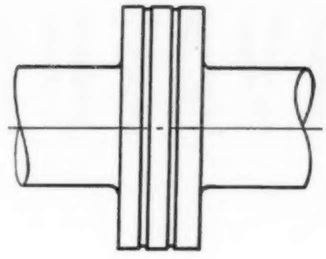


Fig. 134—Use of sealing lands with interrupting circumferential grooves reduces friction loads on valve spools

resistance to three equally spaced depressions on a land; the pressure at each depression will depend on local clearance, increasing as the latter decreases, thus giving self-centering action.

Slide valves and, for that matter, rotary valves as well are subject to hydrodynamic forces caused by throttling of fluid at the ports. These forces arise because the fluid being throttled leaves or enters the spool at an angle, even in the absence of taper at the lands. With perpendicular lands the angle is about 69 degrees, *Fig.* 136. These effects have been extensively analyzed‡ and it is shown that the force is always a centering one, and is given by

$$F = 0.0045q\sqrt{p}$$

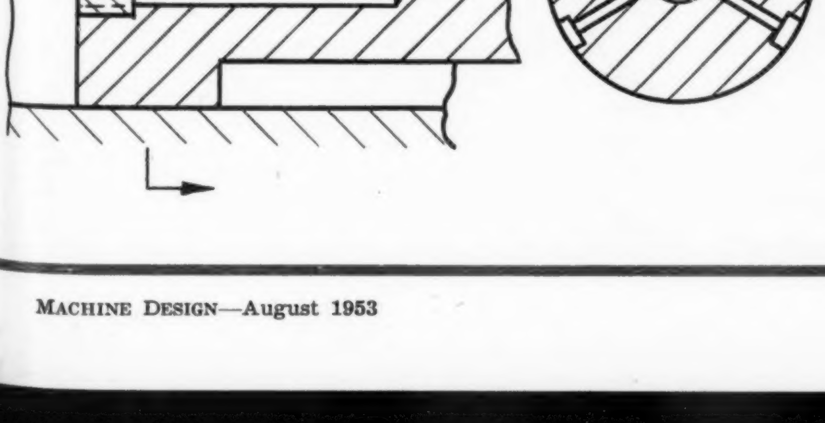
for each port at which throttling takes place, F being the force in pounds, p the pressure drop through the port in psi, and q the flow in cu. in. per sec. This applies to perpendicular lands—in the presence of taper the force in somewhat higher. It is also shown that there are transient forces depending on the rate of movement of the valve, which may in some cases cause instability.

A recommended very rough assumption for design purposes, which is probably on the conservative side for the majority of typical slide valves, is to take the initial friction as 0.03 times the product of the peak system pressure, p, and the cross section area of the sleeve bore, A. On this basis, to give a healthy margin, a return or centering spring should have an initial strength of say $0.05pA$. It is possible to tolerate a substantial decrease in load with spring extension, since friction drops to almost zero as soon as the valve begins to move. On this basis, the load required to operate a spring-centered or spring-returned valve will be $0.08pA$. If the valve is operated by pressure, a minimum load of $0.1pA$ should give an adequate margin.

Similar considerations apply to nominally balanced rotary valves, but assessment of the initial friction seems to be considerably more difficult, and it is doubtful if any simple general rules can be devised.

‡ Shih-Ying Lee and J. F. Blackburn—"Contributions to Hydraulic Control. 1—Steady-State Axial Forces on Control-Valve Pistons. 2—Transient-Flow Forces and Valve Instability," *ASME, Transactions,* 1952.

Fig. 135—Valve spool design with three resistance flow paths to provide self-centering action for low friction



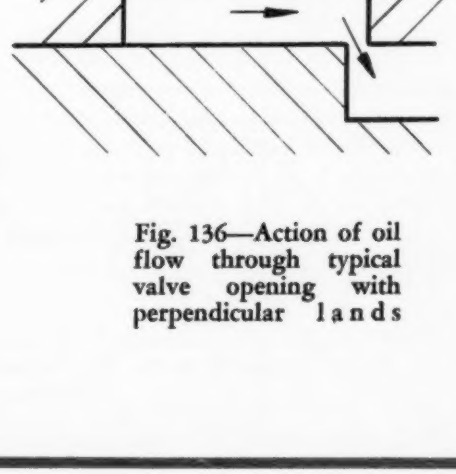


Fig. 136—Action of oil flow through typical valve opening with perpendicular lands

CAM CURVATURE

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PROBLEMS in the design of high-speed cam mechanisms, in which inertia forces and elastic properties of the follower linkage become major factors in the behavior of the system, are currently receiving increased attention. Considerable interest is being shown, for example, in the effects of rate of change of follower acceleration. Cams designed for follower motions based on the parabolic law, those based on the simple harmonic law, and cams machined as a series of circular arcs and tangents are criticized because of the theoretically infinite rates of change of acceleration encountered at the transition points from one curve to another. As a consequence, more complex follower motion laws are being considered; the cycloidal law, in particular, has received much attention. Designs may also be worked out for follower acceleration schedules based on experimental work and not conforming to any simple or readily written motion equation.

An important factor entering any detailed cam analysis, whether the design be based on a simple or a complex motion law for the follower, is the cam curvature. The radius of curvature enters any calculation of contact stress between cam and follower. Knowledge of the relation between cam radius of curvature, follower motion, and cam base circle is useful in selection of the size of the base circle for a proposed design.

Some of the published material on the computation of radii of curvature is listed at the end of this article.[1-4] The most comprehensive treatment is Baxter's. The presentation in Baxter's paper makes use of differential geometry. The purpose of this article

[1]References are tabulated at end of article.

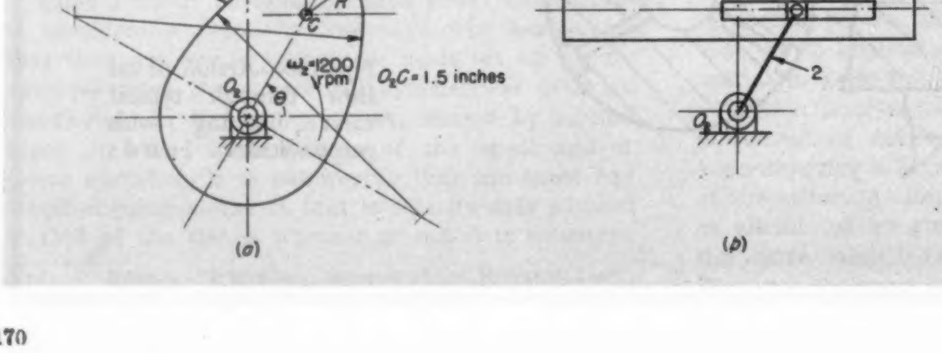


Fig. 1—Cam of known contour with reciprocating flat-faced follower, *a*, and its kinematic equivalent, *b*

GRAPHICAL METHODS for finding velocities and accelerations of cam followers by use of relative velocity and relative acceleration equations are presented in this three-part series. How the same equations can be used to determine the cam radius of curvature for any specified follower motion is also discussed. Specifically treated are reciprocating in-line and inclined flat-faced followers, reciprocating and oscillating roller followers, oscillating flat-faced follower, and reciprocating roller follower with reciprocating cam.

Part 1—Reciprocating In-Line Flat-Faced Follower

is to present a graphical method of approach and, at the same time, to show how the graphical solution may be used to give an analytical equation.

The basic principle underlying this discussion is that geometric properties of cams, such as radius of curvature, can be established from kinematic properties—from velocities, accelerations, etc. In other words, the thinking is done in terms of velocities and accelerations, even though the final result may be a purely geometric quantity. The general idea, usually credited to Hartmann,[4] can be illustrated by reference to any plane curve.

If the curve is given in equation form, either in rectangular or polar co-ordinates, for any point of the curve the radius of curvature can be calculated by well-known formulas developed in elementary calculus. If, on the other hand, one was not given the equation of the curve but was told that a particle traveling the curve had a velocity V and an acceleration A at a certain instant, then for that point of the curve one could find the radius of curvature R from

$$R = V^2/{}_nA \quad (1)$$

where ${}_nA$ is the normal component of acceleration (the component of acceleration perpendicular to the velocity). In the study of machine kinematics, equations for paths of motion are usually very difficult to write and work with, but techniques for determining instantaneous velocities and accelerations are comparatively simple. Hence, while at first glance it may seem a rather roundabout procedure to use velocity and acceleration analyses to determine geometric quantities, actually it proves to be a very useful tool in kinematic design.

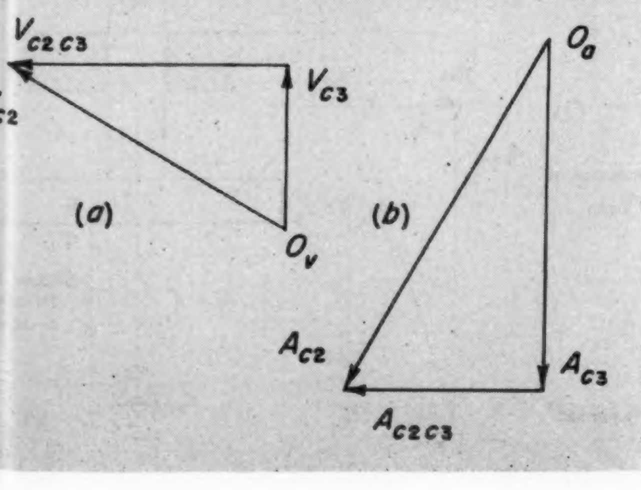
This article will expand on the foregoing idea as it applies to disk cam mechanisms, reviewing methods for determining velocities and accelerations and showing how the same methods can be used to determine cam radii of curvature when motions are specified. Graphical processes will be used. For most work, the numerical results obtained from the graphical solutions are sufficiently accurate. However, it will also be shown that analytic expressions can be developed from the vector diagrams more easily, the authors believe, than by more conventional mathematical methods. One of the major advantages of the graphical method is that it is equally easy to apply regardless of how simple or complex may be the motion specified for the follower.

The cams analyzed in this article will be presented in two ways:

1. To determine the motions for a prescribed cam.
2. To determine the radius of curvature for a prescribed motion.

Although the two types of problems seem quite dif-

Fig. 2—Velocity polygon, *a*, and acceleration polygon, *b*, for systems shown in *Fig.* 1



ferent at first glance, the techniques of handling the problems are very similar, with a simplification of the background necessary for both situations. The problems selected are given to emphasize the principle of approach rather than the numerical solutions or equations.

Reciprocating In-Line Flat-Faced Follower

Follower Motion: *Fig.* 1*a* shows a cam of known contour working with a flat-faced follower. The cam is assumed to be rotating at a constant angular speed of 1200 rpm counterclockwise and, at the instant being considered, point P_2 of the cam is in contact with point P_3 of the follower. The radius of curvature R of the cam at the point of contact is known. The problem is to determine the motion of the follower.

A kinematically equivalent system is shown in *Fig.* 1*b* where the follower is pictured as being extended, while at the center of curvature of the point of contact of the cam and follower a slider is pictured as being introduced to work in the follower. The equivalence may be seen from *Fig.* 1*a,* where point C remains at a constant distance from the flat face of the follower. The equivalence will hold so long as the radius of curvature of the cam remains constant. If the radius of curvature should change after motion of the cam, a new equivalent system must be used. For the position shown, then, *Figs.* 1*a* and 1*b* are such as to give identical velocity and acceleration to the follower for a given motion of the cam.

A velocity analysis is made of the equivalent system relating the motion of two coincident points: point C_2, the center of curvature of the cam at the point of contact, and point C_3, the coincident point of the follower. Distance O_2C is measured as 1.5 inches. By vectorial addition

$$V_{C2} = V_{C3} \mathrel{+\!\!\!\!\rightarrow} V_{C2C3} \qquad (2)$$

where $V_{C2} = (O_2C)\omega_2 = 1.5(1200)(2\pi)/12(60) =$

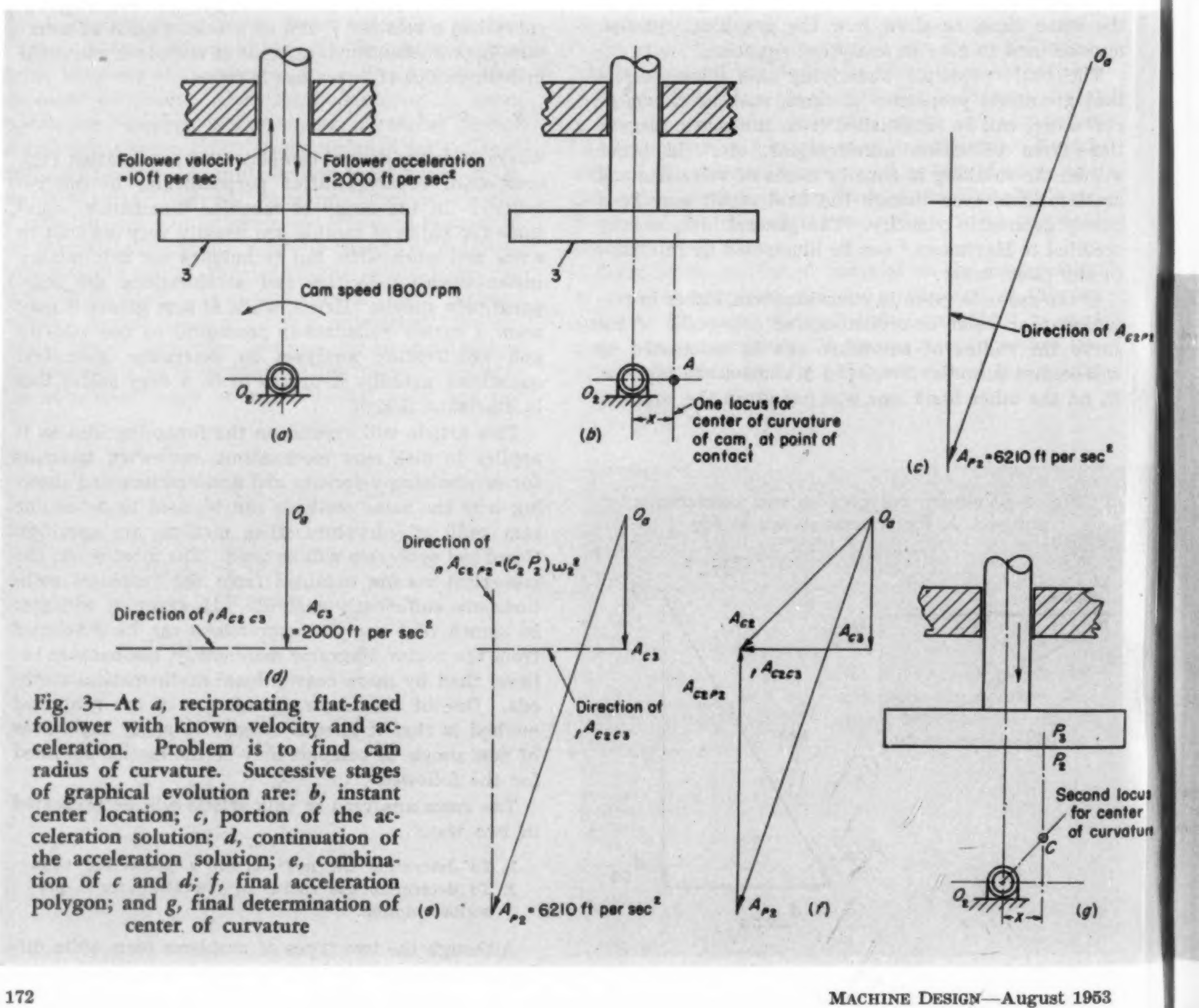


Fig. 3—At *a,* reciprocating flat-faced follower with known velocity and acceleration. Problem is to find cam radius of curvature. Successive stages of graphical evolution are: *b,* instant center location; *c,* portion of the acceleration solution; *d,* continuation of the acceleration solution; *e,* combination of *c* and *d; f,* final acceleration polygon; and *g,* final determination of center of curvature

15.7 ft per sec; V_{C3} = velocity of the follower, known in direction but unknown in magnitude; and V_{C2C3} = relative velocity of the two coincident points C_2 and C_3, known in direction (parallel to the slot in *Fig. 1b*) but unknown in magnitude. *Fig. 2a* shows the velocity polygon.

The acceleration analysis may be made with the kinematically equivalent system. Inasmuch as point C_2 can be considered as a point moving on another moving body, Coriolis' component is indicated. The equation to be used is:

$$A_{C2} = A_{C3} \mathrel{+\!\!\!\!\!\rightarrow} {}_nA_{C2C3} \mathrel{+\!\!\!\!\!\rightarrow} {}_tA_{C2C3} \mathrel{+\!\!\!\!\!\rightarrow} {}_{cor}A_{C2C3} \quad \ldots\ldots\ldots (3)$$

where $A_{C2} = (O_2C)\omega_2^2 = 1.5[1200(2\pi)]^2/12(60)^2$ = 1970 ft per sec² directed from C to O_2; A_{C3} = acceleration of the follower, known in direction but unknown in magnitude; ${}_nA_{C2C3} = 0$; ${}_tA_{C2C3}$ is known in direction (parallel to the slot of *Fig. 1b*) but is unknown in magnitude; and, where ${}_{cor}A_{C2C3} = 2(V_{C2C3})\omega_3 = 0$, since the angular speed of link 3 is zero. Note that point C_2 is considered as the point moving on link 3.

The acceleration polygon in *Fig. 2b* is used to determine the acceleration of link 3 and the relative acceleration of point C_2 with respect to point C_3. For the dimensions selected, the velocity of the follower scales off to 8 ft per sec, and the acceleration of the follower scales off to 1700 ft per sec².

Radius of Curvature: If a cam has been designed for a known motion, where the lift, velocity, and acceleration have been specified and the radius of curvature of the cam is desired, perhaps for the contact stress determination, a velocity and acceleration analysis may be used as an approach to the problem.

CAM CURVATURE

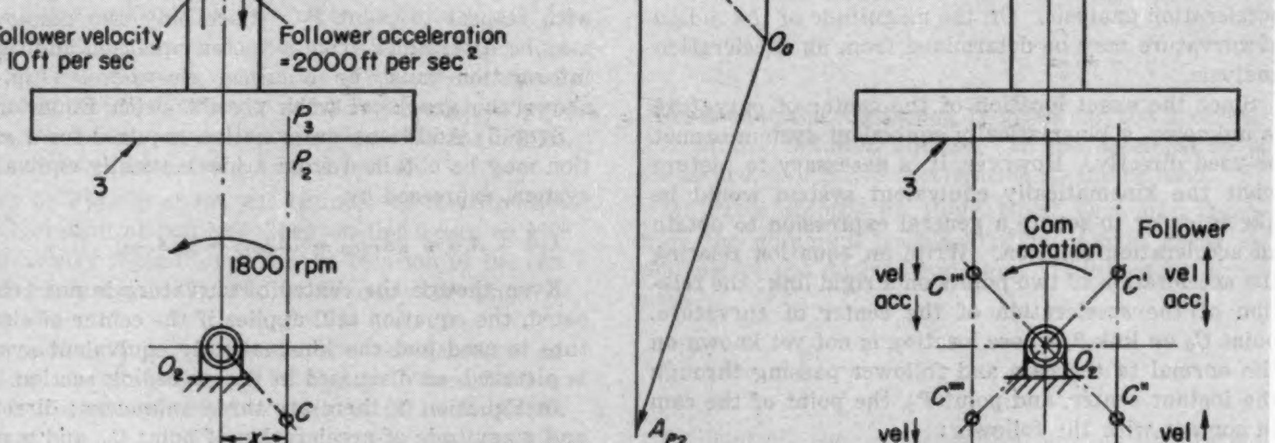
The follower in *Fig. 3a* is known to be in a position where the velocity is 10 ft per sec upward and the follower is accelerating downward at the rate of 2000 ft per sec². The problem is to determine the location of the center of curvature and the magnitude of the radius of curvature to satisfy the prescribed motion at the instant shown for a cam speed of 1800 rpm counterclockwise.

The location of the center of curvature may be obtained graphically by, basically, the determination of two loci. One locus is obtained from a velocity analysis, and the second locus from an acceleration analysis. Or, if the magnitude of the radius of curvature is desired, the magnitude may be obtained from the acceleration analysis. The method of attack as used in the previous problem cannot be used directly inasmuch as the center of curvature is not known. However, the same principles of a kinematically equivalent system will be used to obtain some of the information necessary to solve the equations.

STEP 1: The normal to the relative velocity of the cam and follower at the point of contact is a line on which the center of curvature will lie; or, expressed differently, the common normal to the two bodies at the point of contact, on which the center of curvature must lie, will also include the instant center.

The instant center may be found from $V = x\omega_2$. Then, $10 = x(1800)(2\pi)/60$ or $x = 0.0531$ ft = 0.637 inch where x is the distance from the center of the cam to the instant center, measured along a line parallel to the flat surface of the follower, the line passing

Fig. 4—Case with acceleration and velocity of the follower in the same direction, *a*, and acceleration solution *b*. At *c*, different combinations of velocity and acceleration with corresponding locations of center of curvature



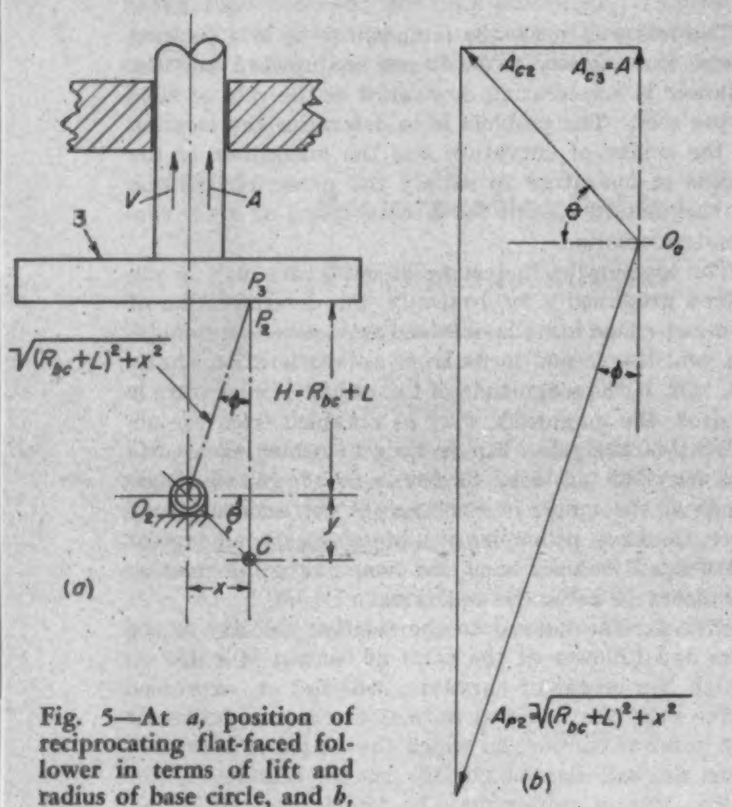


Fig. 5—At *a*, position of reciprocating flat-faced follower in terms of lift and radius of base circle, and *b*, acceleration polygon for reference in development of cam curvature equation

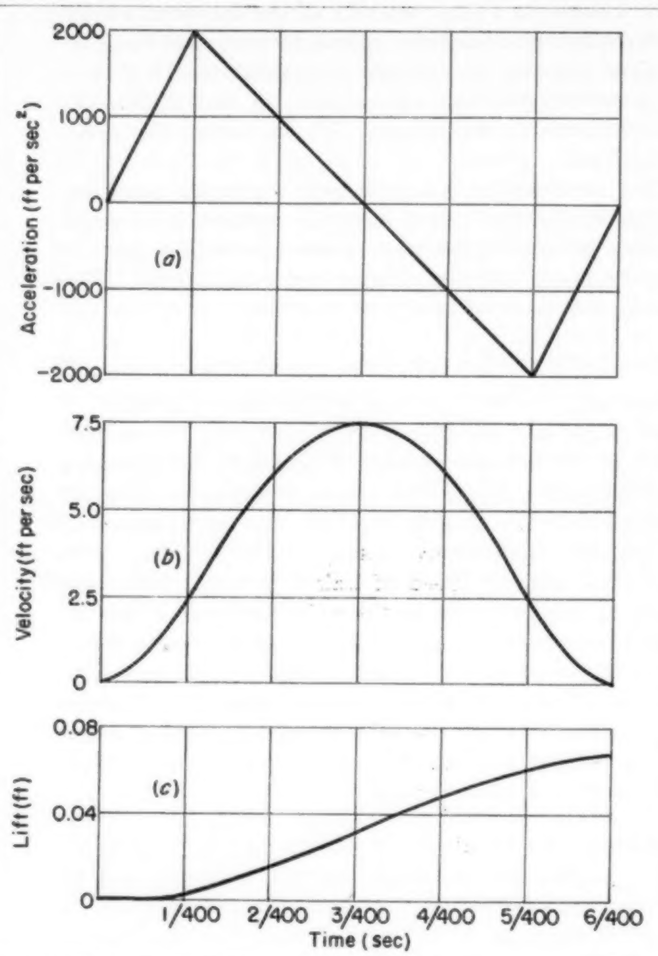


Fig. 6—An arbitrary acceleration curve, *a*, corresponding velocity curve, *b*, and displacement curve, *c*

through the center of the cam. The instant center is given by point A in *Fig.* 3*b*. Point A is located to the right of point O_2 because the cam is rotating counterclockwise and the velocity of the follower is upward. Refer to *Fig.* 4*c* for location of the instant center for a downward velocity of the follower.

STEP 2: A second locus for the graphical determination of the center of curvature is obtained from an acceleration analysis. Or, the magnitude of the radius of curvature may be determined from an acceleration analysis.

Since the exact location of the center of curvature is unknown, a kinematically equivalent system cannot be used directly. However, it is necessary to picture what the kinematically equivalent system would be like in order to set up a general expression to obtain an acceleration solution. Write an equation relating the acceleration of two points on a rigid link: the relation of the acceleration of the center of curvature, point C_2 on link 2, whose location is not yet known on the normal to the cam and follower passing through the instant center, and point P_2, the point of the cam in contact with the follower:

$$A_{C2} = A_{P2} \mathrel{+\!\!\!\!\rightarrow} A_{C2P2} \quad \ldots\ldots\ldots\ldots (4)$$

where $A_{P2} = {}_nA_{P2} = (O_2P_2)\omega_2^2$, a vector directed from P_2 to O_2, which can be determined numerically as $A_{P2} = 2.10[1800(2\pi)]^2/12(60)^2 = 6210$ ft per sec^2; and $A_{C2P2} = {}_nA_{C2P2} = (C_2P_2)\omega_2^2$, a vector known in direction but unknown in magnitude. The direction is the same as the common normal determined in step 1.

Thus, in the equation there are three unknowns: magnitude and direction of acceleration of point C_2 and magnitude of the relative acceleration of point C_2 with respect to point P_2. Since only two unknowns may be determined from a vector equation, additional information must be obtained elsewhere. *Fig.* 3*c* shows the graphical work possible with Equation 4.

STEP 3: Additional information required for a solution may be obtained from a kinematically equivalent system, expressed by

$$A_{C2} = A_{C3} \mathrel{+\!\!\!\!\rightarrow} {}_nA_{C2C3} \mathrel{+\!\!\!\!\rightarrow} {}_tA_{C2C3} \mathrel{+\!\!\!\!\rightarrow} {}_{cor}A_{C2C3} \quad \ldots\ldots\ldots (5)$$

Even though the center of curvature is not yet located, the equation still applies if the center of curvature is used and the kinematically equivalent system is pictured, as discussed in the preceding section.

In Equation 5, there are three unknowns: direction and magnitude of acceleration of point C_2, and magnitude of relative acceleration. The acceleration of the point on the follower, point C_3, is the same as the acceleration of the follower, which has been specified

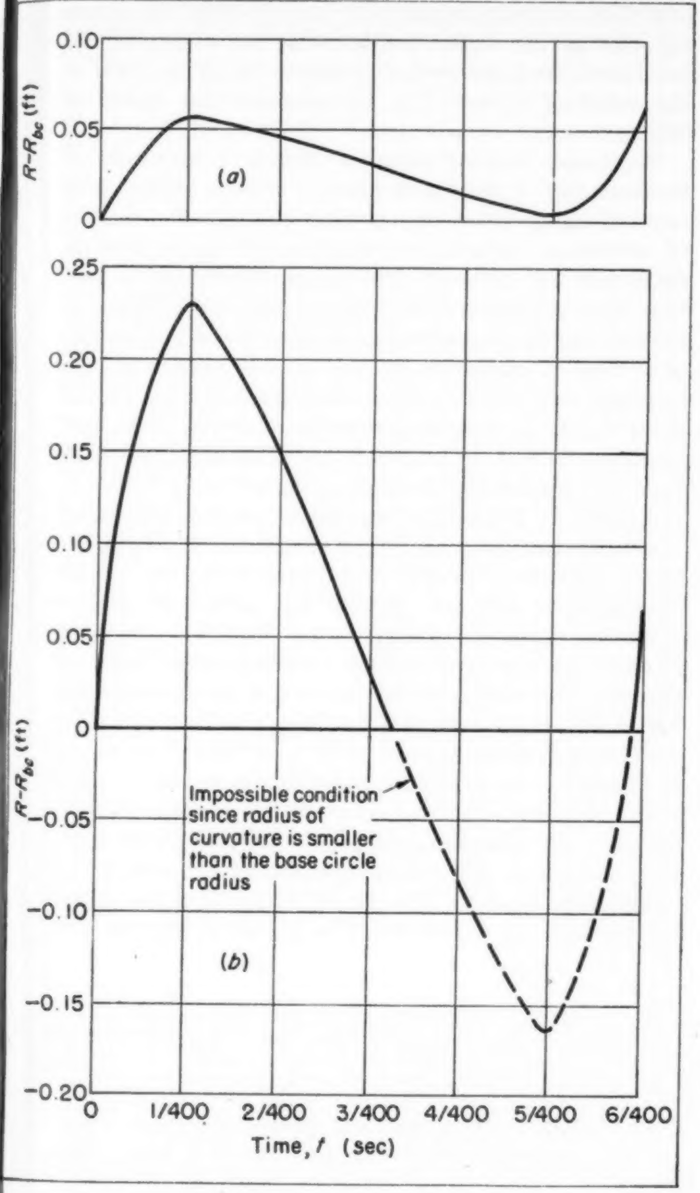


Fig. 7—Difference $R_{bc}-R$ versus time for cam speed of 1800 rpm, *a*, and corresponding curve for 900 rpm, *b*

initially. The Coriolis component is zero. *Fig. 3d* shows the graphical representation of Equation 5.

STEP 4: Equations 4 and 5 may be solved simultaneously to give the acceleration of the center of curvature. *Fig. 3e* shows *Figs. 3c* and *3d* combined. *Fig. 3f* shows the final acceleration polygon, with the magnitude and direction of acceleration of C_2.

STEP 5: *Fig. 3g* shows the transfer of the direction of acceleration of point C_2 back on the figure to give the necessary second locus for the location of the center of curvature. The radius of curvature may be scaled off.

The radius of curvature may be calculated, if desired, by use of: ${}_nA_{C2P2} = (C_2P_2)\omega_2^2$. Then $3930 = (C_2P_2)\ (1800(2\pi)/(60))^2$ or $C_2P_2 = 0.110$ ft $= 1.32$ inches.

OTHER POSSIBILITIES: *Fig. 4a* shows the solution for the case where the velocity and acceleration of the follower are in the same direction, with the position of the follower being known. With the cam speed being given, the distance x can be computed as before. Its value is the same as before for the same cam speed, and for the same velocity of the follower. The acceleration solution is shown in *Fig. 4b*.

Various possible locations are shown in *Fig. 4c* for the center of curvature for different combinations of direction of velocity and acceleration of the follower for a counterclockwise direction of rotation of the cam. These points are determined automatically by the proper velocity and acceleration analysis.

Exact Radius of Curvature: If greater accuracy of cam radius of curvature is desired than is obtained by a graphical solution, which is sufficiently accurate for most purposes, one may use the graphical solution to obtain analytical relations to give as much accuracy as desired.

In *Fig. 5a* the follower is shown moving with a velocity V (positive upward) and with an acceleration A (positive upward). The follower is in a position at a distance of the radius of the base circle plus the lift from the center of rotation of the cam: $H = R_{bc} + L$.

From *Fig. 5a* $\tan\theta = y/x$. From *Fig. 5b*, the acceleration polygon,

$$\tan\theta = \frac{A}{\sqrt{(R_{bc}+L)^2+x^2}\ (\omega_2^2 \sin\phi)}$$

From the properties of *Fig. 5a*,

$$\tan\theta = \frac{A}{\sqrt{(R_{bc}+L)^2+x^2}\ (\omega_2^2)\ \dfrac{x}{\sqrt{(R_{bc}+L)^2+x^2}}}$$

$$= \frac{A}{x\omega_2^2}$$

Therefore

$$\frac{y}{x} = \frac{A}{x\omega_2^2}$$

Since $R_{bc} + L = R - y$,

$$R = R_{bc} + L + \frac{A}{\omega_2^2} \qquad (6)$$

Note that the expression is given for the acceleration of the follower upward. If the acceleration is downward, a negative acceleration should be used.

MINIMUM RADIUS OF CURVATURE: One may determine the minimum possible radius of base circle by analysis of the derived equation. Transposing terms gives

$$R - R_{bc} = L + \frac{A}{\omega_2^2} \qquad (7)$$

To illustrate the procedure of attack, an illustrative problem is selected. *Fig. 6a* shows an arbitrary acceleration curve, so selected as to avoid as much as possible any sudden application of forces to the cam.

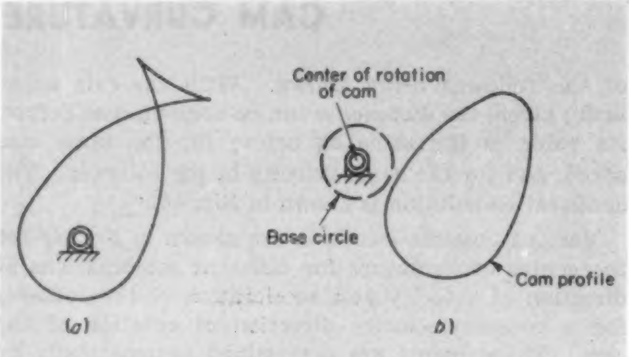


Fig. 8—Negative curvature obtained with too small a base circle, *a*, and cam with negative base-circle radius, *b*

The designer is at liberty to select any type of acceleration curve—the analysis to be presented is general.

Integration of the acceleration curve gives the velocity curve, *Fig. 6b*. Integration of the velocity curve gives the displacement or lift curve, *Fig. 6c*.

Only a portion of the follower motion, the rise of the follower, will be considered, inasmuch as the same procedure to be given may be applied for the rest of the motion of the follower.

In *Fig. 7a* is shown the plot of the difference, $R - R_{bc}$, between the radius of the base circle and the radius of curvature of the cam at the point of contact for a cam speed of 1800 rpm. The difference is determined by calculation of $L + A/\omega_2^2$. *Fig. 7b* shows the plot of the difference between the radius of the base circle and the radius of curvature of the cam at the point of contact for an assumed cam speed of 900 rpm.

The lower limiting value of radius of curvature of the cam with a flat faced follower is zero, with a consequent sharp point on the cam profile. The radius of curvature cannot be negative or else a reverse curvature will occur in the cam, as illustrated in *Fig. 8a*. Such a condition will not permit the follower to have the initially prescribed motion. If the lower limit of radius of curvature of the cam is zero, then the smallest radius of the base circle that can be used is determined by the minimum value of $R - R_{bc}$ for a plot as shown in *Fig. 7a* or *b*. As an illustration, consider *Fig. 7b*, where the minimum value for $R - R_{bc} = -0.1650$ ft. If the limiting value for R is zero, then the minimum value for R_{bc} is 0.1650 ft. Now assume that a minimum radius of curvature desired on the cam is to be 0.5 inch (0.0417 ft). From $R - R_{bc} = -0.1650$, with $R = 0.0417$, $R_{bc} = 0.2067$ ft necessary to give the required motion. At any other point of the cam, the radius of curvature will be greater than 0.0417 ft.

Thus, the minimum radius of base circle will be determined by the minimum point of the plot of $R - R_{bc}$. If the plot of $R - R_{bc}$ should be such that all values are positive, the minimum base circle radius may be negative, for which the cam profile would be outside the base circle, and the cam would rotate about a center not in the cam. Such a cam is shown in *Fig. 8b*.

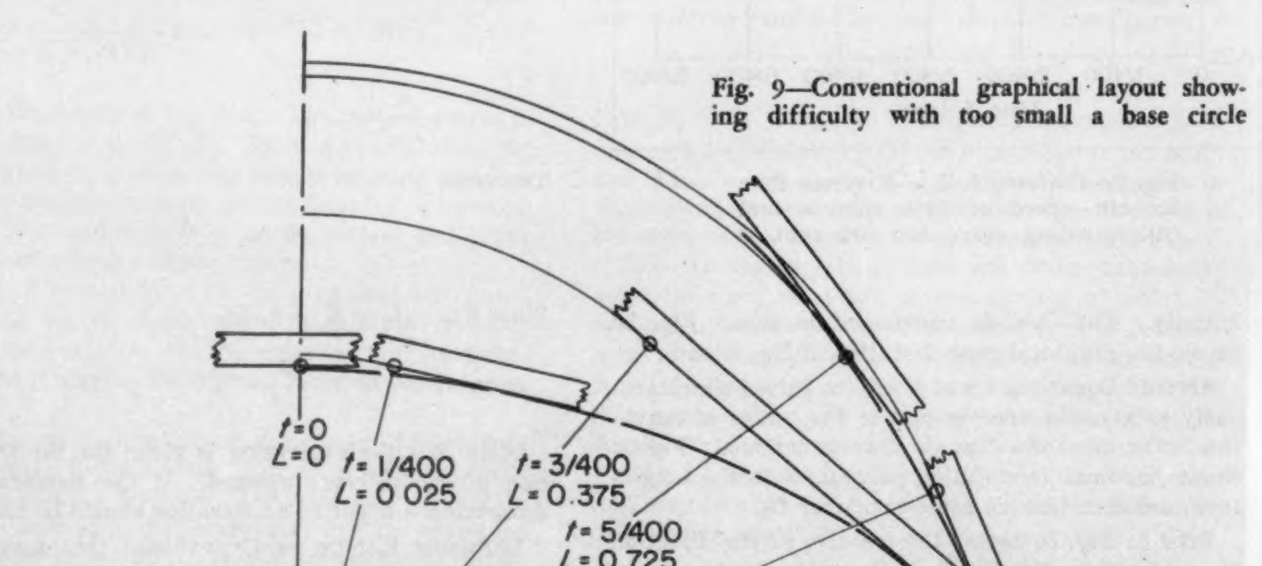


Fig. 9—Conventional graphical layout showing difficulty with too small a base circle

Let us examine *Fig. 7a* for a cam speed of 1800 rpm for the minimum permissible base circle radius. The difference of the radius of the base circle and the radius of curvature is, at $t = 0$, zero, and positive for the rest of the motion. Using the minimum value for $R - R_{bc} = 0$, and setting the minimum value for R as zero, a zero radius of base circle can be used. If an arbitrary value of minimum radius of curvature is selected, say ½ inch, then the minimum radius of the base circle should be ½ inch also, as determined by the conditions at $t = 0$. Note that if the radius of curvature at $t = 0$ is ½ inch, the radius of curvature at any other point will be greater than ½ inch, and positive, for the radius of base circle of ½ inch, as seen from *Fig. 7a*, where all values at times other than $t = 0$ are positive.

A different situation arises in *Fig. 7b*, where the difference of radius of the base circle and the radius of curvature is positive for a range and negative for a range, for a cam speed of 900 rpm for the same prescribed motion as used in *Fig. 7a*. The critical point occurs at $t = 1/80$ second, the minimum value of the curve. The minimum radius of the base circle, assuming an arbitrary minimum radius of curvature permissible anywhere on the cam equal to ½ inch (0.0417 feet), is calculated from $R - R_{bc} = -0.1650$. Then, $0.0417 - R_{bc} = -0.1650$ or, minimum $R_{bc} = 0.2067$ ft $= 2.480$ inches.

Undersize Base Circle: As an illustration of a cam which has too small a base circle, *Fig. 9* is drawn for the acceleration requirements given in *Fig. 6a* for a cam speed of 900 rpm with the base circle taken arbitrarily as 1 inch. The minimum base circle radius found in the preceding section for a minimum cam radius of curvature of ½ inch was 2.480 inches. The cam tangents were determined for $t = 0$, $t = 1/400$ sec, $t = 3/400$ sec, $t = 1/80$ sec, and $t = 3/200$ sec for the lifts given in *Fig. 6c*. Note that if a smooth curve is drawn tangent to the tangent lines, it is impossible to have the cam continuous. The region of difficulty occurs in the vicinity of $t = 1/80$ sec, which is the lowest point of *Fig. 7b*.

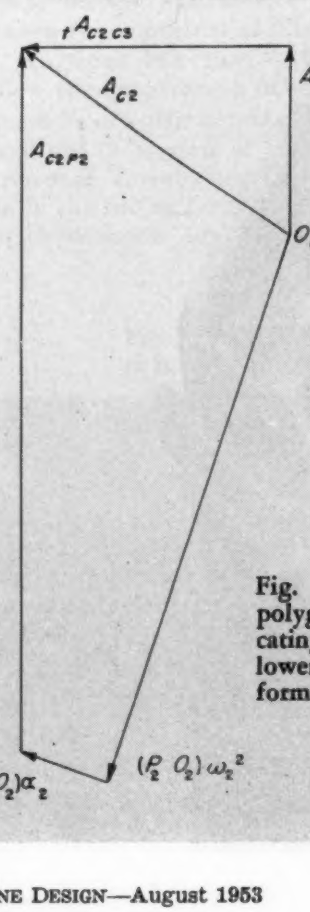


Fig. 10—Acceleration polygon for reciprocating flat-faced follower with nonuniform cam rotation

Nonuniform Cam Speed: The analysis presented is a general one, even though the case analyzed has been for a constant angular speed of the cam. The analysis may be extended, with no increase in difficulty in the solution, to the case where the cam is rotating with an angular acceleration. The acceleration polygon, as well as the equation for radius of curvature, must be modified to take into account the effect of the angular acceleration of the cam. *Fig. 10* has been drawn to show the acceleration polygon for a counterclockwise angular acceleration of the cam. The change from *Fig. 4b* has been the introduction of the tangential component of acceleration: $(P_2O_2)\alpha_2$. It is to be noted that an angular acceleration of the cam will affect the radius of curvature. In this case, the radius of curvature is decreased from the case where the cam is rotating at a constant angular speed. It is to be noted, also, that if the angular acceleration of the cam were taken clockwise, the necessary radius of curvature would have to be increased, compared to the case where the cam is rotating at a constant angular speed.

The equation for radius of curvature, derived from *Figs. 10* and *4a* and presented without any discussion, is

$$R = R_{bc} + L + \frac{VA}{V\omega_2^2 + (R_{bc} + L)\omega_2\,\alpha_2} \quad \ldots\ldots\ldots (8)$$

where R = radius of curvature, ft; L = lift, ft; V = velocity of the follower, ft per sec, considered positive if the follower is moving upward; A = acceleration of follower, ft per sec², considered positive if the follower is accelerating upward; ω_2 = angular speed of the cam, rad per sec, considered positive if the cam is rotating counterclockwise; and α_2 = angular acceleration of the cam, rad per sec², considered positive if the cam is accelerating counterclockwise.

The next part of this series will deal with reciprocating inclined flat-faced and reciprocating roller followers.

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"Research is a process of finding out what you are going to do when you can't keep on doing what you are doing now."—C. F. Kettering.

Deep-Hole Drilling

As a process, deep-hole drilling may offer the designer new opportunities for low-cost drilling of otherwise expensive holes

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PROBABLY the most traditional and best known use of deep-hole drills is for the drilling of gun barrels, *Fig.* 1. Today modern drills made either from high-speed steel or from tool steel with cemented carbide inserts are doing more than just barrel drilling. They are in daily use by automotive, aircraft, farm equipment, machine tool, arms and other manufacturers.

Drills: In general deep-hole drills are highly efficient and versatile tools applicable in many ways to modern manufacture. They are semicustom built, and are produced to exacting tolerances and to customers' specifications. The drill tips are ground to a standard specification upon leaving the factory, but different types of materials or applications may require an alternate pitch, rake, etc. ***Fig.*** 2 shows a standard drill.

Deep-hole drills are being successfully used in two methods of operation, by rotating the work, and by holding the work stationary and rotating the drill. The most advisable and satisfactory, particulary in drill-

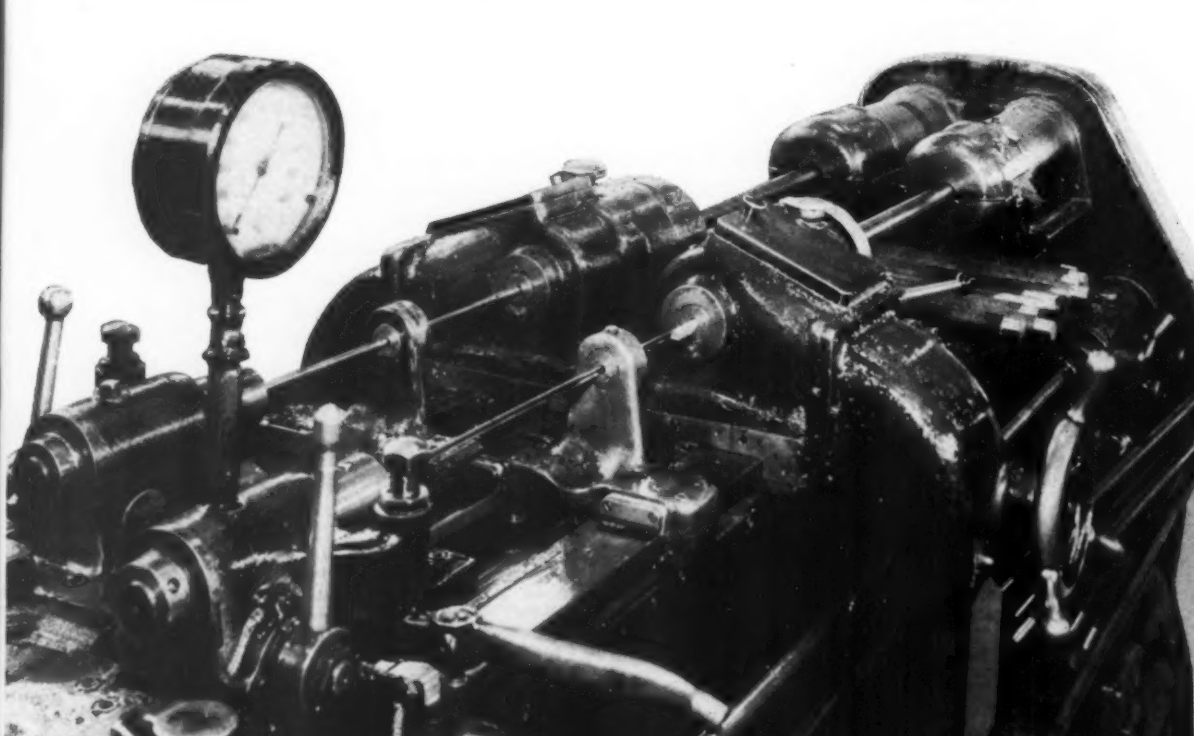


Fig. 1 — Deep-hole drilling a .30-caliber rifle barrel of 4137 steel

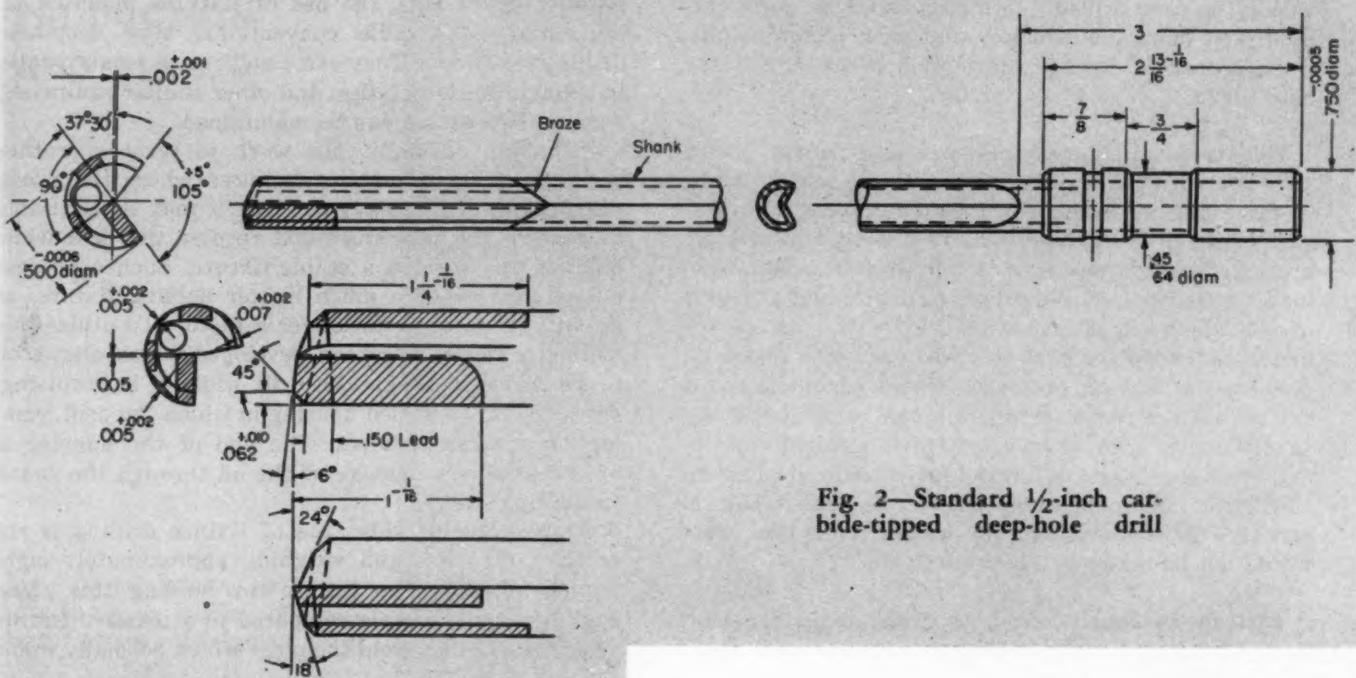


Fig. 2—Standard ½-inch carbide-tipped deep-hole drill

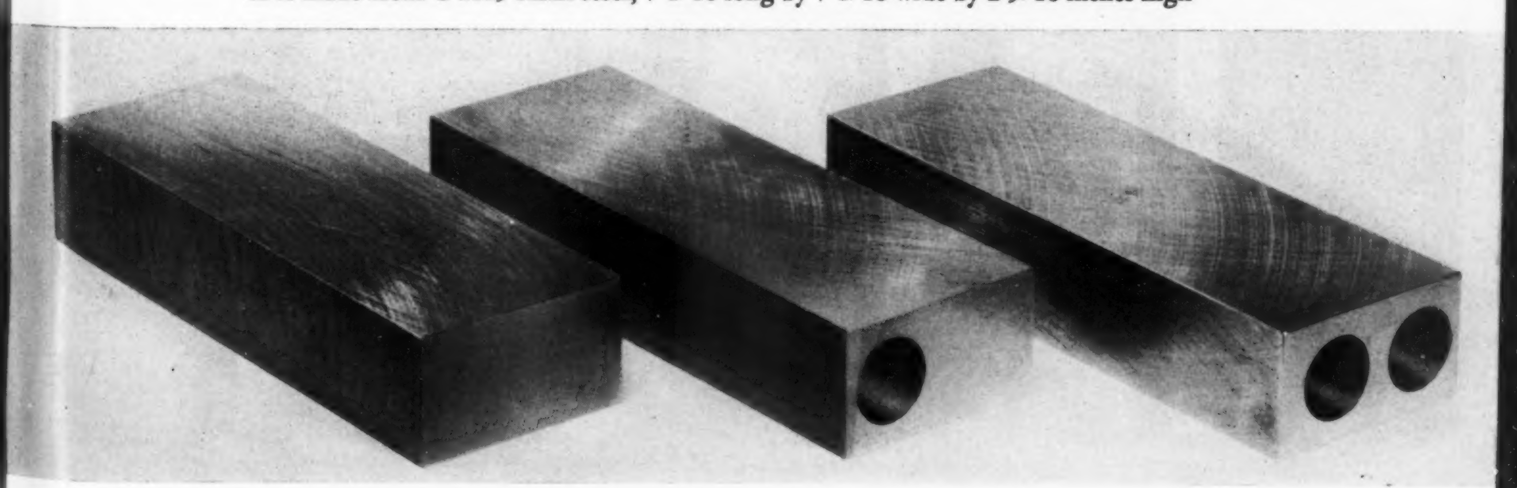
ing to depths greater than 6 inches, is to rotate the work. With the drill in a stationary position, the coolant has a better opportunity to carry the chips back along the V-shank section.

Drilling Range: Best application of deep-hole drills is between the diameters of 0.156-inch through 1.750-inch. Anything less than 0.156-inch diameter does not allow enough strength in the shank. The coolant hole must be of sufficient size to allow adequate coolant, and the V section of sufficient depth to allow chip removal, thereby limiting the allowable cross-section in the tubing for strength. Sizes greater than 1.750-inch diameter are better handled with multi-point cutters, or trepanning type drilling tools.

Depth of hole possible to drill with deep-hole drills is limited only by the capacity of the equipment available. In the drilling of shanks for deep-hole drills over 1-inch diameter, solid bars up to 60-inches length are drilled through their full length by drilling from each end and meeting in the center of the bar.

Hole Finish: Surface finish resultant from this method of drilling will vary with the feed and speed being used, the material being drilled, and the amount of pressure available on the coolant. Mild stock can be drilled at 1800 rpm with a 7 inch per minute feed, 750 psi coolant pressure with a resultant finish of 64 rms or better. Coolant pressure is a large factor in this, since clearing of chips from the cutting edge and guiding strips is necessary to prevent galling or cold welding.

Fig. 3—Shotgun receiver in blank and bored form. Weighing 8 pounds, the receiver is made from C-1119 ARR steel, 7 1/16 long by 7 1/16 wide by 2 9/16 inches high



Materials: Any material that can be successfully handled with conventional methods of drilling, can usually be deep drilled. In many instances, parts can be drilled more economically and with better results when proper set-up and application is made of deep-hole drills.

Tolerances: Diameter tolerance and runout, as in conventional drilling, will vary with the requirements of the job being done. Hard spots encountered in the path of the drill will tend to make it walk. However, in properly treated materials, a tolerance of plus 0.002-inch can be realized, regardless of depth, and a runout of 0.001-inch per 3 inches of depth. It must be remembered, however, that this will vary with material, feed, speed, and oil pressure. Where runout is not a prime factor, greater production can be realized. As in drilling of gun barrels, where a straight hole is essential the barrel is straightened after drilling to the bore. In applications where the drilled hole is strictly for oil passage or weight reduction, this would not be necessary of course.

Fixture Drilling: Deep-hole drills are also being used advantageously for boring holes through many parts other than gun barrels in the manufacture of shotguns, rifles and pistols. These holes are economically drilled with the use of fixtures mounted on horizontal two-spindle conventional type deep-hole drilling machines. They can also be used on automatic indexing machines, lathes and other similar equipment where oil pressures can be maintained.

Although, normally, the work is revolved rather than the drill, in certain instances where the workpiece is especially heavy or large it may be advisable to reverse the procedure and revolve the drill while holding the work in a stable fixture. Such a process allows the use of a much lighter holding fixture, as centrifugal force is no longer a factor. To utilize this method a device has been developed which allows oil to be forced through the drill while it is revolving. Oil is fed into a sealed housing in which the drill bushing or adapter revolves. The end of the adapter is open and allows passage of the oil through the shank to the tip.

An example of this type of fixture drilling is the receiver of a shotgun weighing approximately eight pounds, *Fig.* 3. The fixture for holding this piece, *Fig.* 4, is quite simple compared to a massive fixture required to safely hold the piece which normally would

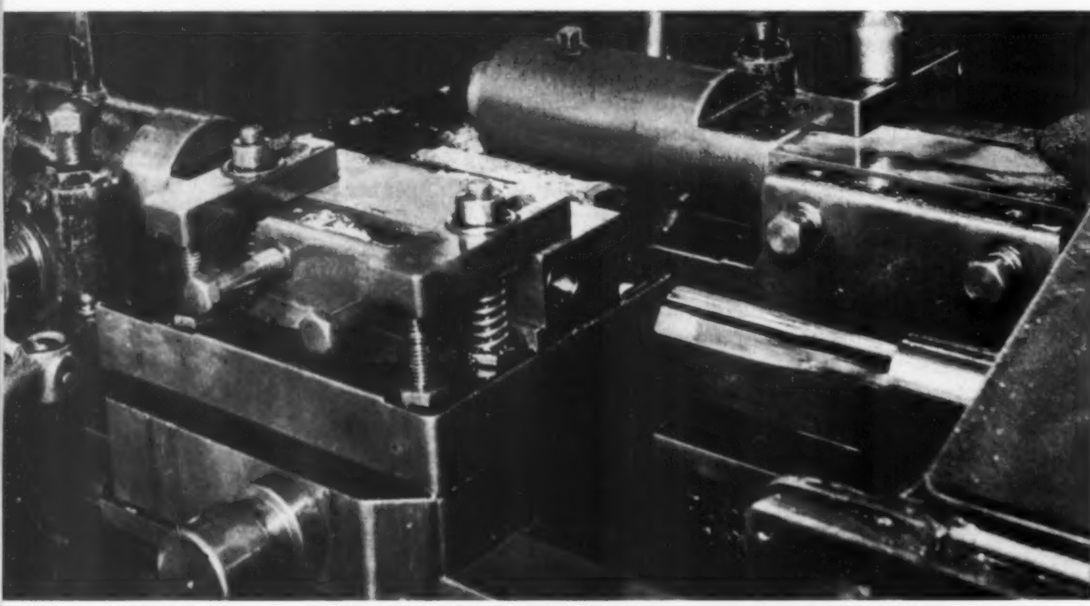
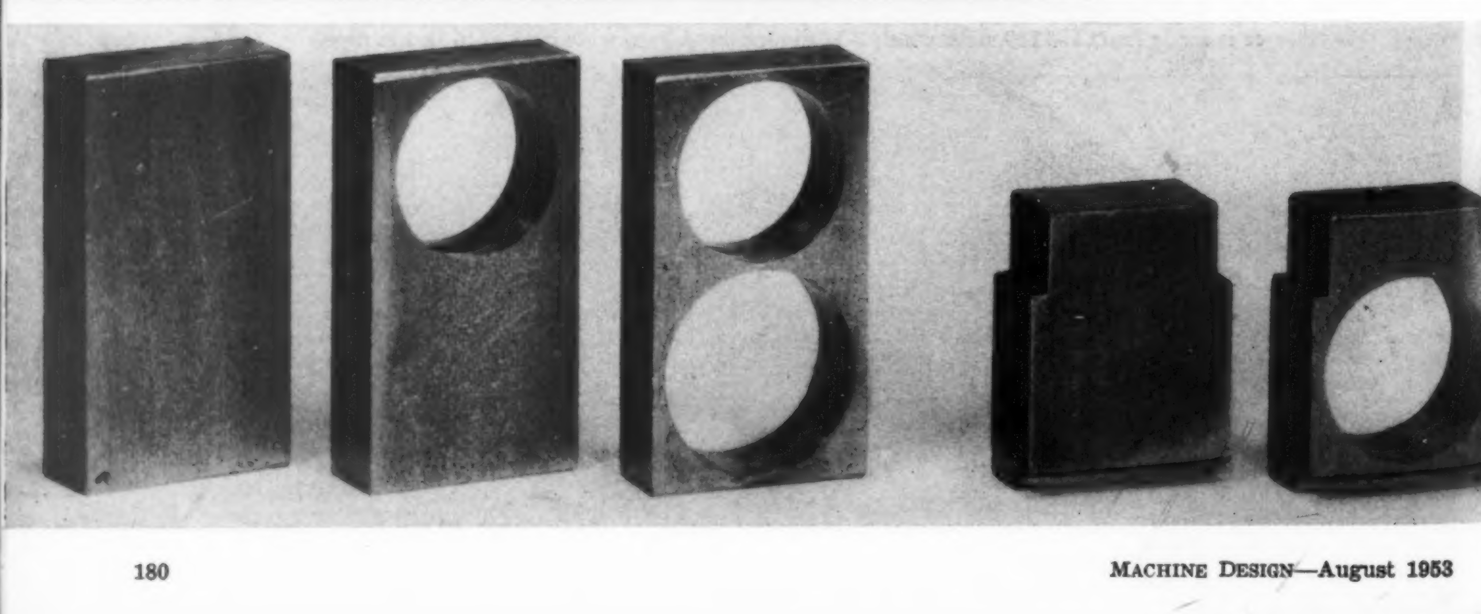


Fig. 4—Left—Deep-hole drilling arrangement for the receiver of *Fig.* 3 with the drill rotating in lieu of the fixture

Fig. 5—Below—Group of typical parts, with holes, suitable for gang drilling in fixtures



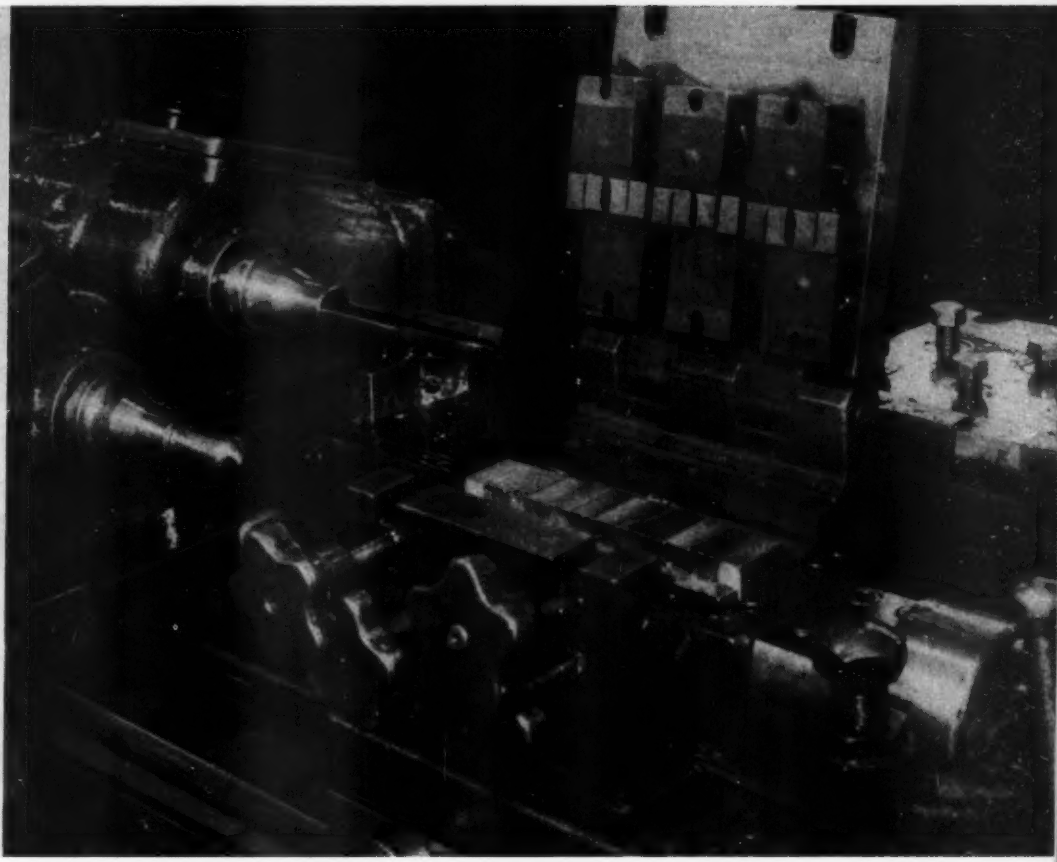


Fig. 6—Stationary fixture for gang drilling a large group of similar parts

revolve at 1900 rpm. Two holes are drilled into the end of the piece; one, through the block, the other, 6½ inches into the block as a blind hole.

The deep-hole drill in this case is of the carbide insert type, 0.890-inch diameter, with two guide wear strips and a cutting strip. The drill is fed into the piece at 1½ inches per minute with oil at 650 pounds pressure and the drill revolving at 1900 rpm. The run-out in an operation of this kind is less than 0.006-inch.

Advantages of drilling in this manner are: (1) holes can be drilled faster than with other methods as the drill does not have to be withdrawn to clear chips, (2) the cutting edge is kept cool with a constant flow of oil, and (3) chips are washed back out through the V in the drill and shank under pressure to assure a hole free from rings and score marks.

Another variation of deep-hole drilling is called gang drilling in fixtures. This is a method of stacking several small pieces and drilling through the group as a unit. Gang drilling is made practical by the inherent accuracy and tendency of a deep hole drill to run true. A dozen or more pieces of the type shown in Fig. 5 which call for through holes, may be loaded into a fixture and drilled in one operation. This reduces handling time considerably and adds to the efficiency of the operation. The pieces to be drilled, shown in *Fig.* 5, have been cut from raw stock, machine rolled to remove burrs, and set as a gang into the fixture, *Fig.* 6. The stock in each case is C-1119 ARR and is drilled through by a 0.892-inch carbide tipped drill fed at 1½ inches per minute with an oil pressure of 650 pounds. The drill in both cases revolves at 1900 rpm.

In the two cases described in the preceding paragraphs the drill used is of the carbide tipped type where greater speed and longer life of the drill itself is demanded. The wear strips are longer than the cutting strip so that the entire length of the cutting strip can be used with complete drilling accuracy while at the same time maintaining quality in the operation. In drilling most mild steels, however, high-speed steel tips can be used to good advantage.

Economy Considerations: Oil pressures are an important factor in deep-hole fixture drilling but most horizontal and vertical machines are capable of supplying the correct quantity of oil under the proper pressure. High Standard uses a central oil system for a battery of machines and pressures are regulated for each individual job. A central system is not necessary, however, as single pumping units fixed to machines can produce the desired results, but the use of a central system often results in a more economical operation.

Deep-hole drills can be used with greater economy in practically any application, whether it be a through or blind hole. The only applications not suitable are where intermittent cuts are encountered, or on cored holes. Either of these would require special and probably prohibitive costly set-up.

Deep-hole drills are economical as the shanks and bushings may be returned to the factory for retipping. If a tip is worn it need not mean complete loss of the drill. In many other ways cost reductions can be made through the use of deep-hole drills if they are tied in with modern manufacturing methods. Engineers should be aware of the inherent possibilities of this little-known method.

Acceleration Testing Centrifuge

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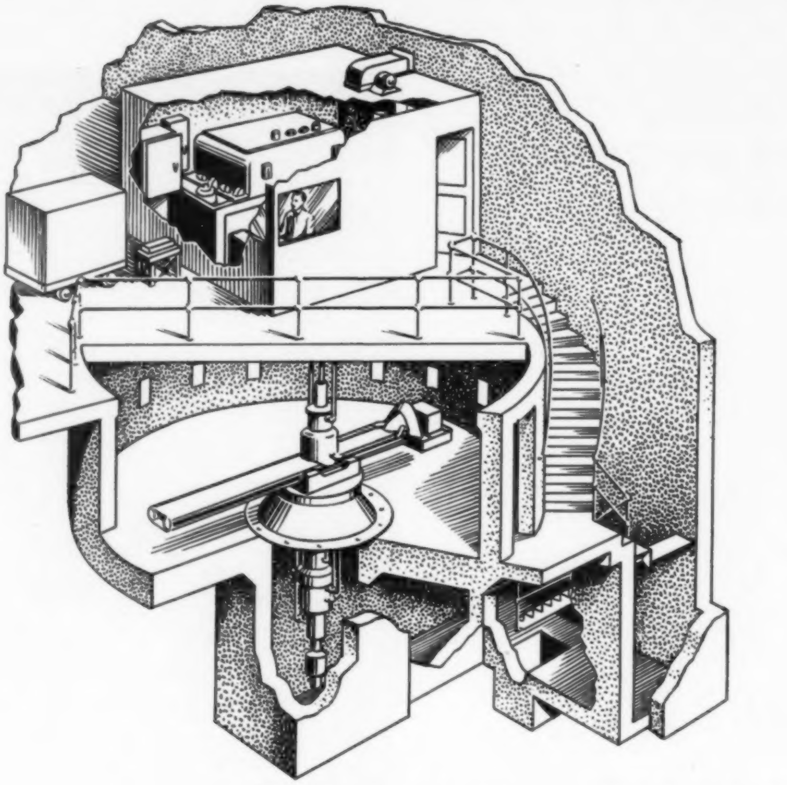


Fig. 1—Rotor pit, below ground, is ballasted with 250 tons of concrete. Controls are on upper level

OPERATIONAL tests of components subjected to high acceleration loading when in use, such as guided missile components, can be done quickly and easily using the large centrifuge described in this article. Equipment to be tested may have a maximum length of 6 feet and maximum weight of 600 pounds. Maximum acceleration of the unit is 65*g* at a rotational speed of 160 rpm.

Since the centrifuge is located in a pit, *Fig.* 1, all portions of which are not visible from the control room, precautions must be taken to prevent injury to persons working in the pit due to accidental starting of the centrifuge. The control system, therefore, incorporates a number of safeguards.

The main pump unit which supplies the hydraulic drive motor, can be started only if the controls are in neutral—in other words, if there is no possibility of starting the rotor. Presuming that the rotor controls are set at neutral, the main power unit can be energized and a small pressure built up in the system; then when the operator desires to energize either of the rotors, he presses the "start" button. At this time, the pits are warned by a Klaxon which blows for approximately 15 seconds—this period being adjustable.

During this time, if a technician is within a danger area, he can kill the main power unit by merely hitting a knockout button which breaks the electrical circuit and requires the operator at the control console to reset the whole machine before it can be startd again. This procedure can be handled time after time if the operator is not aware that he is being forced to reset his panel by someone in the danger area. However, 15 seconds is considered ample time to allow anyone to leave the danger area and let the test proceed or notify the control unit, through the two-way communication system that has been set up.

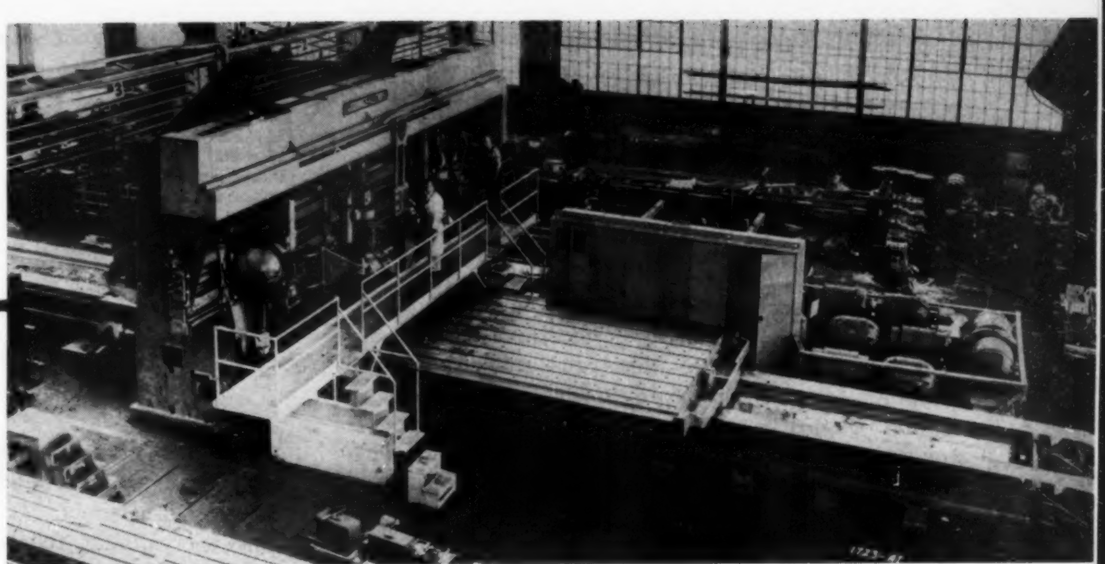
After the Klaxon has stopped, a ready light on the control console lights and the operator then has complete control of acceleration and speed. A timing device permits setting of a predetermined length of run, so a test can be run for a given length of time and automatically stopped. In stopping, either manually or automatically, the fluid motor that drives the rotor functions as a pump and, by restriction of the output, effectively brakes the rotor within at least 60 seconds.

If this system should fail for any reason, a mechanical brake which is electrically energized immediately stops the rotor. This brake is designed to bring the centrifuge under control from top speed at top loading within 30 seconds. The mechanical unit is only used in case of emergency.

One of the interesting design features is the number of leads required to obtain all data from the test specimen, namely:

1. Forty signal leads with shielding to eliminate electrical interference, noise or cross talk.
2. Three heavy-duty, 50-ampere service slip rings for conducting power into the instrument platform on the end of the rotor.
3. Three hydraulic leads, capable of taking hydraulic oil at a pressure of approximately 1500 psi to the instrument platform, and a motor-pump system mounted directly on the rotor to return the oil to the center of the rotor against the effects of centrifugal force.
4. A 1000-psi air joint to conduct an accurately measured volume of high-pressure air to different pieces of equipment which are air driven.
5. A microwave waveguide as well as coaxial joints for leading signals into certain parts of the electronic system.

Fig. 1—Giddings and Lewis Hypro skin-milling machine equipped with multiple-disk clutches for machining whole wing or fuselage sections of jet fighters



Pros and Cons of *Hydraulically Actuated Clutches*

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FOR a number of years, oil-applied clutches have been designed and manufactured for special applications but standardized clutches of this type have become available only recently. They are finding increasing application because of several advantages that result with the use of hydraulic pressure as the direct clamping medium. But at the same time hydraulic actuation introduces certain features that may be disadvantageous in relation to mechanical actuation. As a preliminary guide in the selection and application of hydraulically applied clutches, this article presents a summary of their characteristics.

This discussion is based upon multiple-disk clutches of the type in common use in machine tools, *Fig.* 1, and in automatic industrial transmissions. However, in principle the points to be mentioned hold in general for any clutch form. Comparison of mechanical and hydraulic actuation means is provided by the cutaway views, *Figs.* 2 and 3. Basically the clutches are quite similar, but the oil-applied type incorporates an integral oil cylinder to clamp the plate stack.

Advantages: The most obvious advantage with the oil-actuated clutch is that no adjustment is required to compensate for the friction plate wear, since the floating or pressure plate is the ram of the cylinder. As the plate stack wears, the ram travel increases automatically.

This design feature provides a correlative advantage. Torque capacity of the clutch for any given oil pressure is always a constant. Constant pressure is not held in a mechanically actuated over-center type clutch which requires adjustment. Each clutch engagement, creating plate wear, causes the torque value to decrease. Normally, this decrease is a small value per engagement, but in the case of high energy pickups, the decrease can be relatively great. In any event, if the clutch is not adjusted periodically, the torque capacity will eventually drop to the load value. On the next engagement the clutch will fail to pick up the load, with the result that the clutch will continue to slip. If slippage is not detected, the clutch will soon burn up. More clutch failures can be attributed to lack of adjustment than any other cause.

A second correlative advantage is that with constant torque capacity assured, a smaller clutch may often be used. That is, because a mechanically actuated clutch does not have a constant torque capacity, selection of a clutch larger than required for the actual starting load is often necessary, even after this load has been corrected by an applicable service

factor. The clutch engineer or machine designer does this by applying factors, gained from experience, to each installation—judging the rate of plate wear likely to be experienced, the type or "mechanical aptitude" of the machine operator, and the accessibility and location of the clutch relative to the operator.

Torque capacity of any clutch is the product of clamping force, the mean radius of the plate stack, the number of rubbing surfaces, and the friction coefficient. Consequently the torque capacity of the oil-applied clutch is directly proportional to the applied oil pressure since this pressure creates the clamping force. Therefore torque capacity is a variable quantity readily controllable by the operator if a suitable hydraulic system is employed. This characteristic is not generally required, but there are certain installations where controlled torque is advantageous.

Substitution of the oil-applied clutch for the conventional mechanically actuated clutch may show cost savings under certain conditions. Economy can

Table 1—Comparative Clutch Data

	Clutch Disk Diameter (in.) 3	4	5	6	7
Mechanically Actuated Multiple-Disk (Duplex—to run in oil)					
Working Torque (lb-in.)	480	1600	2925	5800	8800
Power per 100 rpm (hp)	0.76	2.5	4.6	9.2	14.0
Maximum Power (hp)	9	20	36	57	78
Maximum Speed (rpm)	6000	4600	3700	3200	2800
Hydraulically Actuated Multiple-Disk (Single—to run in oil)					
Working Torque (lb-in.)*	480	1540	2830	5700	8450
Power per 100 rpm (hp)	0.76	2.4	4.5	9.0	13.4
Maximum Power (hp)	9	20	36	57	78
Maximum Speed (rpm)	2500	2500	2000	2000	2000

* Based on minimum oil pressure of 150 psi.

be gained if the present clutch is remotely actuated by air or hydraulic cylinder, or electrical means such as a solenoid. If a suitable source of oil pressure is available, the oil-actuated clutch will require a selector valve and shaft-around-seal or an end-joint seal, depending upon the nature of the installation. The cost of these items must be balanced usually against a throwout fork, operating shaft, and hand lever.

Another source of cost saving is reduced machine downtime since no clutch adjustment is needed. Also, the wear life of the oil-actuated clutch is greater in most applications because less clutch slippage occurs since full design torque is always available.

Disadvantages: The mechanically actuated clutch will hardly be made obsolete by its companion, the oil-actuated clutch. The latter has one great primary limitation: a source of suitable hydraulic pressure must be available. The bulk of clutch applications today are on machines which are not as yet provided with a hydraulic system or on one which does not provide adequate pressure for clutch actuation.

A second limitation which is not generally realized is that very little "clutch feel" can be obtained by oil actuation. Clutch feel is an ambiguous term but is meant to indicate the sense of touch the operator experiences as he engages a clutch. He soon learns to correlate this touch to rate of engagement. The oil-applied clutch lends itself best to those applications where a positive rapid engagement is required. Where a need exists for controlling the rate at which the clamp force is applied during the engagement cycle itself, the mechanically actuated clutch, manually operated, provides optimum performance.

A certain amount of feel can be obtained with oil-applied clutches in installations where the nature and production volume of the application warrant special valve design. This feature is generally limited to industrial "power-shift" transmissions where the use justifies a fairly complex special valve which may easily cost more than the clutch.

However, there are relatively few applications where clutch feel is necessary. In most instances the driven shaft simply needs to be brought up to speed, and the rate of engagement is not important. Too often a designer attempts to gain cushioned acceleration through clutch slippage at the expense of plate life when soft acceleration could be better gained with no wear by a hydraulic coupling used in conjunction with the clutch.

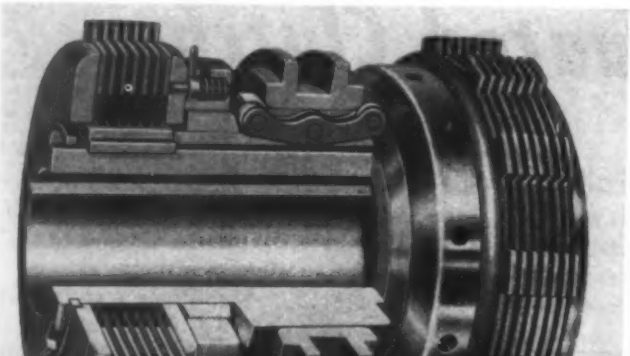
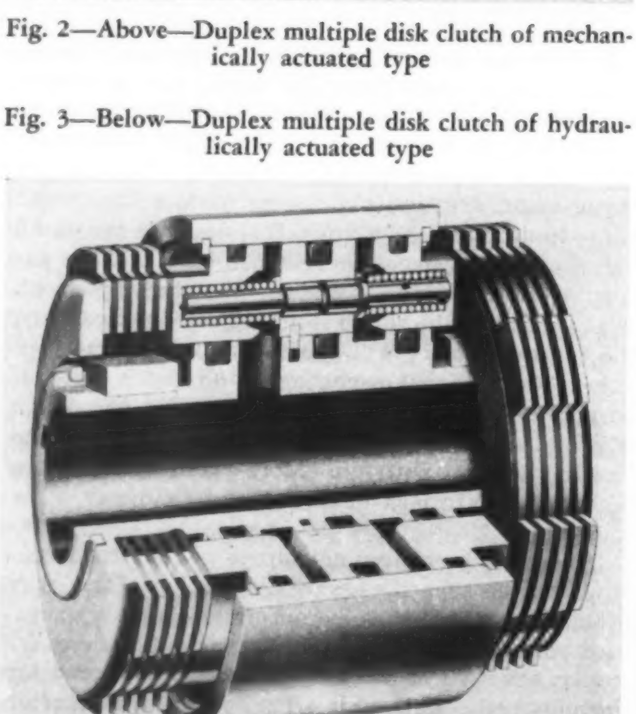


Fig. 2—Above—Duplex multiple disk clutch of mechanically actuated type

Fig. 3—Below—Duplex multiple disk clutch of hydraulically actuated type



Evaluating MOMENTS OF INERTIA by Graphical Techniques

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IN DESIGN, strength or vibration analysis of unsymmetrical structural cross-sections is frequently encountered. One difficulty with such profiles is that simple formulas are not available for the computation of their inertia. A typical example might be the turbine blade shown in *Fig.* 1. Its profile, although determined by nonstructural considerations, demands a rigorous structural analysis. For such an analysis, the major and minor principal moments of inertia must be evaluated.

Aimed at avoiding complicated mathematical treatments in the structural analysis of nonsymmetrical profiles, this data sheet presents the following simple analysis techniques:

1. An experimental method for determining the center of gravity.
2. A graphical method for determining moments of inertia about arbitrary axes drawn through the center of gravity.
3. A graphical method for determining the major and minor moments of inertia from three arbitrary moments of inertia.

Center of Gravity: The center of gravity can be as-

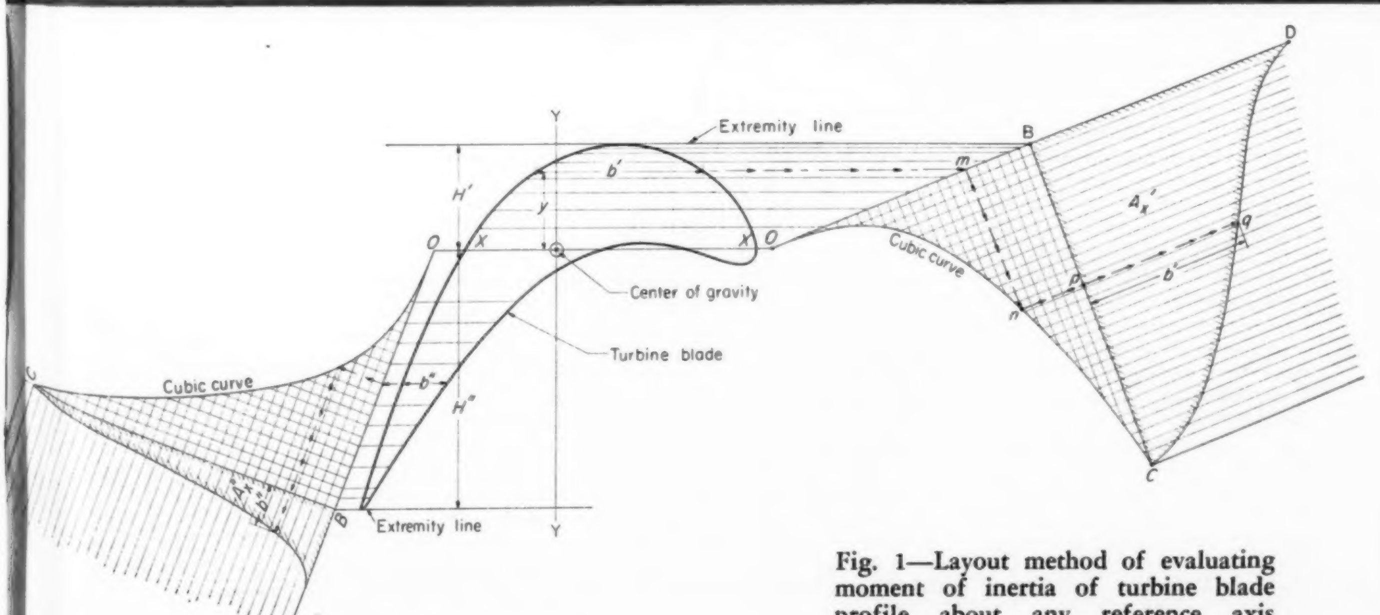


Fig. 1—Layout method of evaluating moment of inertia of turbine blade profile about any reference axis

DATA SHEET

certained experimentally from a cardboard cutout of the subject profile. The profile should be scaled-up in size enough to make the major dimension of the cross-section about 10 inches long. This model of the profile is then suspended from a pin driven through an extremity of the cross-section. A plumb line drawn through the pin runs through the unknown center of gravity. Repeating this procedure with a pin located at some distance around the periphery from the first point will establish a second plumb line. The intersection of these lines marks the center of gravity. To check this point, three or four plumb lines should be drawn.

Moment of Inertia: In the subsequent discussion, a basic mathematical integral is given for determining the moment of inertia about a reference axis of a nonsymmetrical profile. A graphical method for evaluating this expression is outlined and a proof of the validity of the method is presented.

MOMENT OF INERTIA INTEGRAL: The mathematical definition of the moment of inertia about any reference axis, *XX*, is

$$I_x = \int_{-H''}^{H'} b\, y^2\, dy \qquad (1)$$

Graphically this integral can be depicted as the area of the curve of by^2 plotted against y where b is the length of each line parallel to the reference axis as shown in *Fig.* 1 and y is the perpendicular distance measured from the reference axis to any stratum line b. To make a plot of such a curve requires tedious repetitive computations which can be eliminated by employing a graph of the cubic curve $(BC)/(OB)^3 l^3$ versus l as shown in *Fig.* 2.

CUBIC GRAPH CONSTRUCTION: For the cubic graph, the length *OB* of the abscissa should be made slightly longer than the longest distance, measured from the reference axis to the profile extremities, that is likely to be encountered in any computations. The maximum ordinate *BC* of the cubic curve may be freely chosen.

For flexibility in working with different profiles, curves such as shown in *Fig.* 2 should be made in a variety of suitable sizes and a quantity of prints of each kept on hand. Sets of three sizes based on *OB* equal to 2, 5, and 10 inches are suitable for most applications. However analysis problems encountered will readily determine practical sizes to stock.

GRAPHICAL METHOD: The technique can be easily demonstrated by an analysis of the turbine blade profile shown scaled-up *S* times in *Fig.* 1. The center of gravity can be determined by the experimental method already presented. The moment of inertia of the profile about the arbitrary *XX*-axis can be determined by

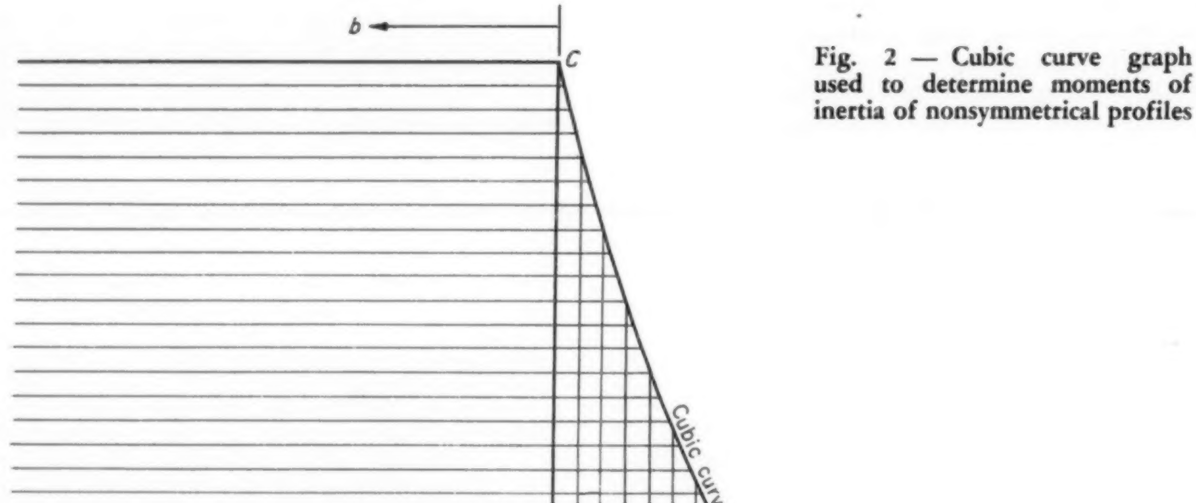


Fig. 2 — Cubic curve graph used to determine moments of inertia of nonsymmetrical profiles

a method which utilizes the cubic graph. In this approach the area moments of inertia for the areas of the profile on each side of the XX-axis are determined individually. The total moment of inertia I_x is the sum of the two values or

$$I_x = I_x' + I_x'' \quad (2)$$

where I_x' is the moment of inertia for the upper portion of the profile and I_x'' is the moment of inertia for the lower portion. The moment of inertia of the profile rotated about any other axis through the center of gravity can be similarly established.

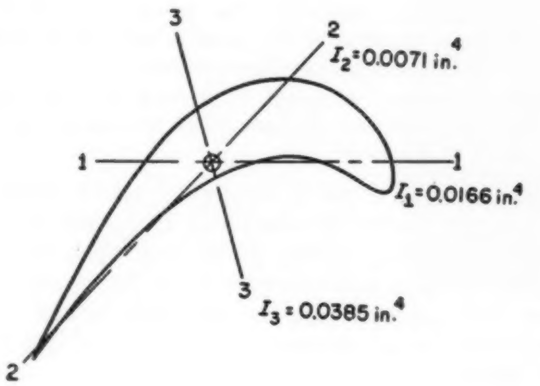


Fig. 3—Turbine blade profile showing three arbitrary axis lines through the center of gravity and moments of inertia values about each axis

The moment of inertia for an area of the profile on one side of an axis is determined by the following steps. This procedure assumes that the reference axis has been drawn through the center of gravity of the profile.

1. Lay off a line parallel to the reference axis and tangent at the extremity of that half of the profile under analysis.
2. Place a suitable size print of the cubic graph shown in *Fig.* 1 adjacent to the profile such that point O of leg OB lies on the reference axis line and point B lies on the extremity line. Leg OB must always be made longer than H, the distance between the reference axis and the parallel extremity line.
3. At any stratum y project a line parallel to the XX axis until it intercepts the leg OB at point m.
4. Run a line perpendicular to leg OB from point m to point n on the cubic curve, and at right angles continue from point n along the graph lines to point p on leg BC.
5. With dividers transfer the total length b' at stratum y to the graph $OBCO$, placing it perpendicular to leg BC (from point p to point q).
6. Repeat this procedure at other y strata between the reference axis and the parallel extremity line.
7. Draw a smooth curve CqD through the ends of the perpendiculars erected on the leg BC.
8. Determine the area within curve $BpCqDB$ by a planimeter or by geometrical means.
9. Determine the moment of inertia from

$$I_x = A_x' \frac{(H')^3}{3(BC)} \quad (3)$$

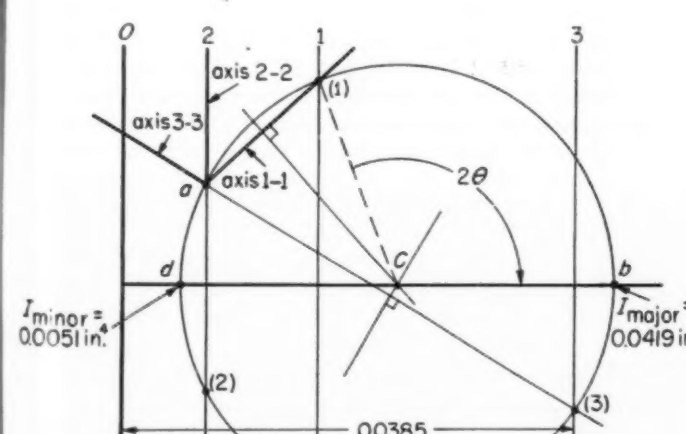


Fig. 4—Mohr's moment of inertia circle for determining location and magnitude of principal moments of inertia

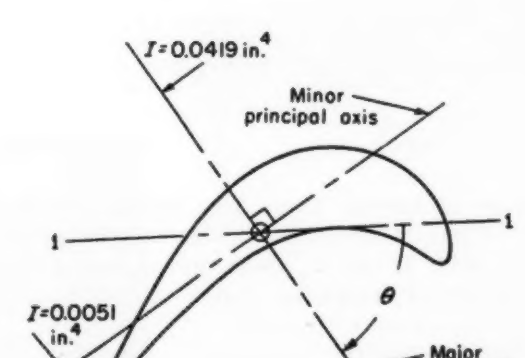


Fig. 5—Turbine blade profile with principal moments of inertia axes located and evaluated from Mohr's inertia circle

Equation 3 has been developed from

$$A_x = \int_B^C b\, d\left[\frac{(BC)l^3}{(H')^3}\right] = \frac{3(BC)}{(H')^3}\int_B^C bl^2 dl$$

Since ratio y/H' is equal to the ratio $l/(OB)$,

$$l = \frac{(OB)y}{H'},\ l^2 = \frac{(OB)^2 y^2}{(H')^2},\ \text{and } dl = \frac{(OB)}{H'}dy$$

Hence

$$A_x' = \frac{3(BC)}{(OB)^3}\frac{(OB)^3}{(H')^3}\int_0^{H'} by^2\, dy \qquad (4)$$

But since

$$I_x' = \int_0^{H'} b\, y^2\, d\, y$$

Equation 4 resolves into Equation 3.

This moment of inertia determination is made from a profile that is scaled-up S times in size. This scale factor S is the ratio of drawing size to actual size. The actual value of the moment of inertia is found by dividing the results of Equation 3 by S^4 or

$$I_x' = \frac{A_x'\,(H')^3}{3(BC)\,(S)^4} \qquad (5)$$

EXAMPLE: Known factors in a numerical analysis of the turbine blade profile shown in *Fig*. 1 are H' = 1.9 inches, H'' = 4.68 inches, (BC) = 6.25 inches, and S = 6. Employing the graphical technique as outlined in the foregoing procedure, A_x' = 17.32 square inches and A_x'' = 2.73 square inches. From Equations 2 and 5

$$I_x = \frac{A_x'\,(H')^3}{3(BC)\,(S)^4} + \frac{A_x''\,(H'')^3}{3(BC)\,(S)^4}$$
$$= \frac{17.32(1.9)^3 + 2.73(4.68)^3}{3(6.25)\,(6)^4}$$
$$= 0.01655 \text{ in.}^4$$

Transfer of Inertia Axes: The relationship of I_θ, moment of inertia about an axis inclined θ degrees to axis XX, with I_x and I_y, moments of inertia about axes XX and YY, respectively, is

$$I_\theta = \frac{(I_x + I_y)}{2} + \frac{(I_x - I_y)\cos 2\theta}{2} + J_{xy}\sin(-2\theta)$$

where J_{xy} is the product of inertia, and θ is the angular location of the major principal moment of inertia relative to a reference axis. The form of this expression is similar to that encountered in the transfer of stress axes. In that application the formula is often resolved into a graphical solution by the use of Mohr's circle.

The following are the properties of the Mohr's circle for moments of inertia.

1. Moments of inertia are measured along the horizontal axis.
2. Products of inertia are measured along the vertical axis.
3. The co-ordinates (I_x, J_{xy}) establish one point on the circle, and the co-ordinates $(I_y, -J_{xy})$ establish another point on the circle diametrically opposite.
4. The intersection of the horizontal axis and the diagonal (I_x, J_{xy})-$(I_y, -J_{xy})$ is the center of the circle.
5. The moment of inertia properties about an axis θ degrees away from a reference axis are determined by the co-ordinates on the circle established by rotating a diameter 2θ degrees from the reference diameter. The sense of rotation is the same as that of the relative rotation between the new axis and the reference axis.
6. The principal moments of inertia, I_{maj} and I_{min}, are the extreme dimensions of the circle on the horizontal axis.

As discussed in the foregoing the inertia circle can be determined by first computing I_x, I_y, and J_{xy}. Actually J_{xy} is not readily evaluated. The following method permits the construction of the inertia circle from three arbitrary moment of inertia values. These may be two perpendicular values such as I_x and I_y and any other third value I, or they can be any arbitrary relationships such as I_1, I_2, and I_3 which are oriented at convenient angles of inclination as in *Fig*. 3. The procedure is:

1. In *Fig*. 4 the vertical line OO represents a zero moment of inertia value. From this line measure off distances horizontally which will represent I_1, I_2, and I_3 values as determined graphically by employing the cubic curve method. Construct vertical lines 1-1, 2-2, and 3-3 at these points.
2. Superpose the axes of *Fig*. 3 on any point a of line 2-2 such that axis 2-2 of *Fig*. 3 is colinear with vertical line 2-2 of *Fig*. 4. It is important that *Fig*. 3 is superposed and not its mirror image.
3. Extend the superposed axes until each axis intercepts its respective line of *Fig*. 4; that is, axis 1-1 intercepts line 1-1 establishing co-ordinates of the circle at point (1). Similarly axis 2-2 intercepts line 2-2 to establish co-ordinates of point (2), and axis 3-3 intercepts line 3-3 at point (3).
4. Erect perpendicular bisectors of lines a-(1) and a-(3). The intersection of these bisectors at c is the center of the inertia circle.
5. Scribe a circle about c through points (1) and (3).
6. A horizontal line drawn through c intersects the circle at two points, b and d, which establish the major and minor principal moments of inertia of the system, I_{maj} and I_{min}, respectively.
7. Angle $(1)cb$ is equal to twice the angle between axis 1-1 and the I_{maj} axis. The actual angle is measured in the direction as shown on the circle in *Fig*. 4.

Accuracy of this method was tested by checking the moment of inertia of a circle. The error observed was less than 3 per cent.

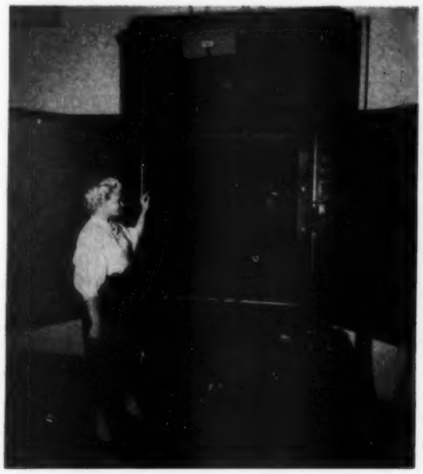
Engineering NEWS ROUNDUP

Big Camera Aids Aircraft Production

A 29-foot long template camera, one of the largest in the aircraft industry, is helping to speed up engineering of delta-wing interceptors and other aircraft at the San Diego Div. of Convair. Accuracy of reproduction of the camera permits reproduction of drawings 12 feet long with an error of 0.002-inch or less. Such accuracy permits drawing and lofting to be done in any convenient scale; drawings can then be reproduced in sizes more convenient for production.

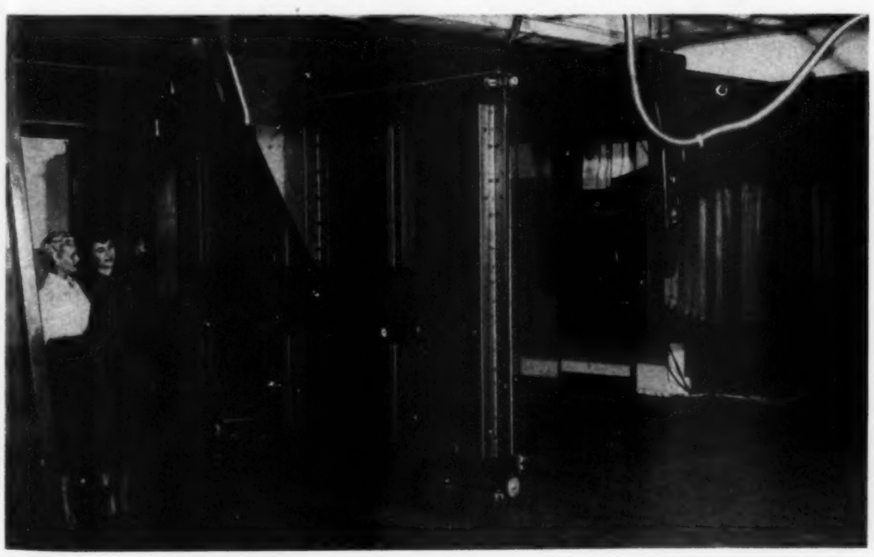
Total bellows extension of the camera is 12 feet. Film up to 42 by 52 inches may be used, and enlargements to 4:1 or reductions of as much as 13:1 can be made. Copy is held on the 12 by 5 foot copyboard by vacuum. Steel sheets weighing as much as 200 pounds can be held firmly in position on the board.

One example of the camera's ability to save time and expense is its use in making shrink reproductions to a certain oversize for casting patterns. Prior to the construction of the camera, it was necessary to make two drawings, one for the finished casting and one for the pattern.



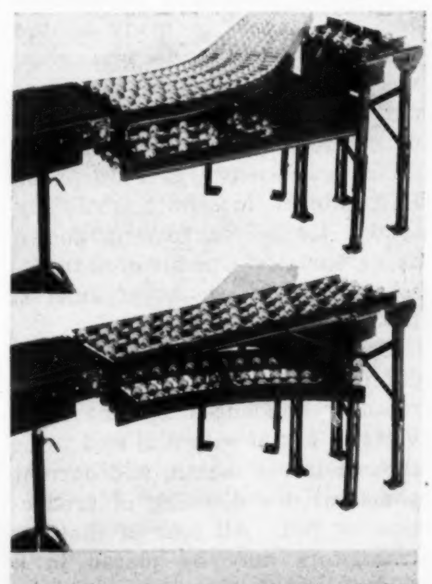
Back of Convair copying camera is in a darkroom. The film holder, like the copyboard, holds the film by vacuum

A periscope on the back of the copyboard permits the operator to watch indicating dials above the camera while adjusting position of the board. Two vacuum lines provide vacuum for holding copy in position



Transistor Developments Indicate Commercial Use Soon

Various recent developments indicate that widespread commercial use of transistors may be a thing of the not too distant future. Some of the factors which have limited the use of transistors to date, such as high cost and scarcity of germanium from which many transistors are made, inability to withstand high temperatures, limited power handling capabilities, and poor performance at high frequencies, are



CONVEYOR SWITCH: Called the Flip Switch, a new wheel-conveyor switch can be rotated to connect a two-way belt conveyor with either of two wheel conveyor lines set at different levels. The entire assembly is pivoted at both ends to permit rapid switching. An automatic stop holds back stock while switching is taking place

INVERTED SPEAKER: The loudspeaker shown is said to incorporate the first significant change in speaker design in 25 years. Placement of the speaker magnet on a spider inside the cone reduces bulk, making large-speaker tone possible in small radios. Carbonneau Industries Inc. will manufacture the speaker exclusively for Motorola

well on the way to being overcome.

Inherent advantages of transistors, which will do many of the things now done by vacuum tubes, are small size, low power consumption and reliability. A testimonial to the reliability of transistors is their application by Bell Telephone Laboratories Inc. in a switching system for toll call dialing known as a translator. One hundred translators were in use as of June 1, 1953.

Commercial availability of four different types of transistors was recently announced by the RCA Victor. Initial output is said to be thousands per month, and current plans call for doubling of production by fall. All four of the new transistors may be placed in a thimble together.

Experimental work done by the same organization has produced transistors which have functioned at frequencies of 425 megacycles per second, well into the ultra high-frequency region utilized by UHF television and a wide range of special electronic communications and industrial equipment. The highest frequency attained by transistors a year ago was 225 megacycles per second.

A new method for producing germanium promises to lower its cost and thereby speed development and commercial availability of transistor devices. Though still in the laboratory stage, the method developed by the General Electric Research Laboratory, produces as many as 100 layers of the special germanium in a six-inch ingot. Other methods are said to produce only one or two layers.

Pure silicon, another transistor material, is now being manufactured by Du Pont. Price of the new material is placed at two to four cents per transistor with all production being done at a small pilot plant. Silicon is attractive as a transistor material because of its ability to withstand temperatures to 400 F. Higher power can thus be handled by a silicon transistor. Additionally, silicon is an extremely plentiful element, in contrast to germanium.

All indications are that transistors will soon be plentiful and economical in addition to their other long-established good qualities. Such a condition will result not only in portable, battery powered television sets, but also in small, compact and reliable electronic machine controls.

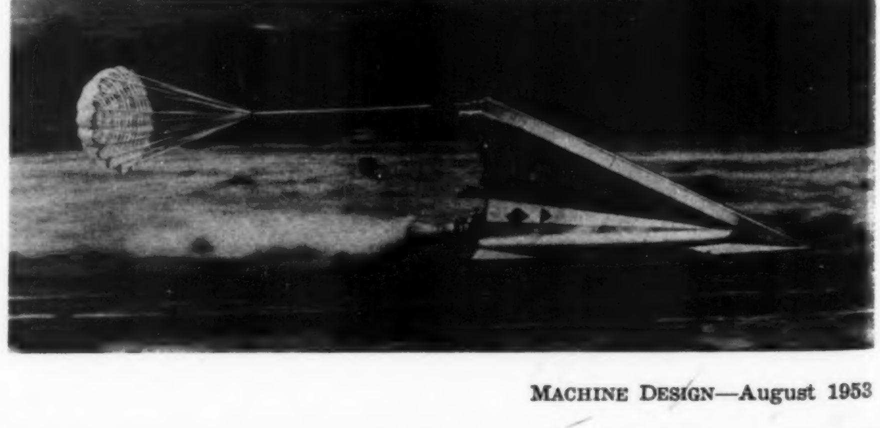
Future is Bright for Engineering Students

Recent campaigns to reduce the critical shortage of engineers are paying off. Through efforts of engineering societies, industries, universities and high school science teachers, a 29 per cent increase in the number of college and university freshmen enrolled in engineering courses in 1952 was registered.

Keeping the high school science curriculum up to date is of primary importance in guiding young people to careers in the technical field. Also playing an important role is industry, making it easier for high school graduates to obtain college educations by offering scholarships and other forms of aid. Part of the flow of new scholarships is due to industry's growing demand for scientists and engineers.

At the Illinois Institute of Technology, 441 students (nearly a quarter of the full-time enrollment) obtained scholarships totaling $264,000. Under the Co-op plan, offered in mechanical, metallurgical, industrial and electrical engineering at I. I. T., students alternate semesters of study with

SUPERSONIC SLED: Capable of attaining speeds of 1500 mph, this rocket-powered sled is held to its guide rails by slippers. Powered by a 50,000-pound thrust North American rocket motor, the test vehicle was developed by Cook Research Laboratories for studing performance of parachute brakes in the transonic and supersonic speed ranges. Although now being used at Edwards Air Force Base for parachute brake study, it may be used for other supersonic testing

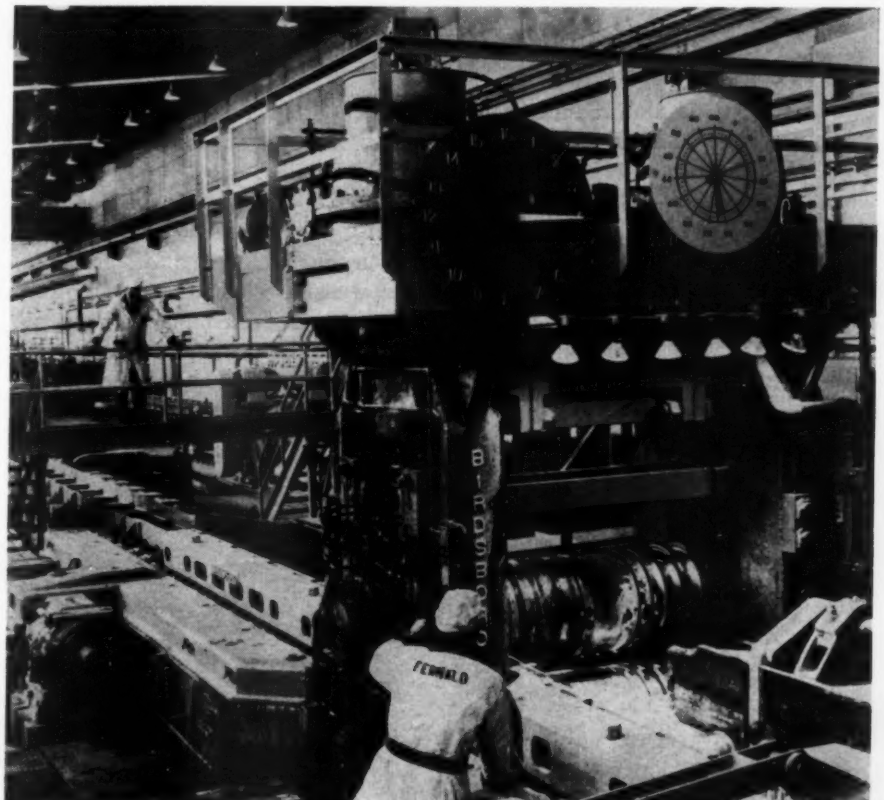


semesters of work. Some industries in this plan also pay the student's tuition and other school expenses.

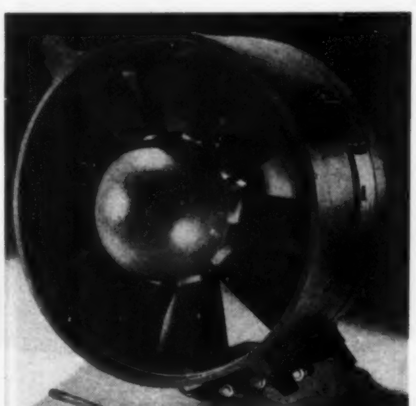
American industry is now hiring one engineer to every 60 production workers. This demand translates itself into spectacular salary advantages for engineers. From his choice of about a dozen jobs, the engineering graduate can expect a starting salary of approximately $80 per week.

Nylon gears are being used in the transmissions of Packard cars. The gears used are a drive gear and pinions for the transmission governor and the speedometer. This is said to be the first time plastics have been used successfully in the complete gear train.

Seventh Annual Summer Meeting of the Wisconsin Society of Professional Engineers will be held at Elkhart Lake, Wis., on September 18 and 19.



URANIUM ROLLING MILL: First mill in this country designed expressly for the production rolling of uranium turns out bars for further fabrication into slugs used in nuclear reactors. Designed and built by Birdsboro Steel Foundry & Machine Co., the mill is part of the Atomic Energy Commission's new uranium production center at Fernald, O., where uranium is produced for use in AEC fissionable materials plants, throughout the country. This center is operated by the National Lead Co. of Ohio



RAM AIR TURBINE: Emergency hydraulic power for operation of aircraft control systems is provided by an air turbine driven hydraulic pump which will supply 3.4 hydraulic horsepower at 130 knots airspeed at outlet-pressures to 3000 psi. Made by AiResearch Mfg. Co., the unit will operate at airspeeds equivalent to Mach 1 or the speed of sound

Advocates "Horse Sense" For Better Engineering

Replacing the "ivory tower perfectionism" of some engineering attitudes with "old-fashioned horse sense" can multiply the effectiveness of American industry's technical manpower, says George M. Muschamp, vice president in charge of engineering of Brown Instruments Div., Minneapolis-Honeywell Regulator Co. Forsaking engineering technique in some cases for simpler, more economical methods would keep projects within the framework of "what will it cost and how long will it take," Mr. Muschamp told the manufacturing conference of the American Management Association in New York City recently.

Good administrative handling can add to the productivity of the now scarce engineer, and the recommendation was made that engineers be taught better evaluation and target-setting habits to reconcile their technical viewpoints with the goals of their employers.

Technical situations must be tempered by "plain economics," said Mr. Muschamp. He suggested that management foster a high delegation of responsibility and an even higher degree of self-management at lower levels.

Honeycomb Construction Used for Dive Brake Door

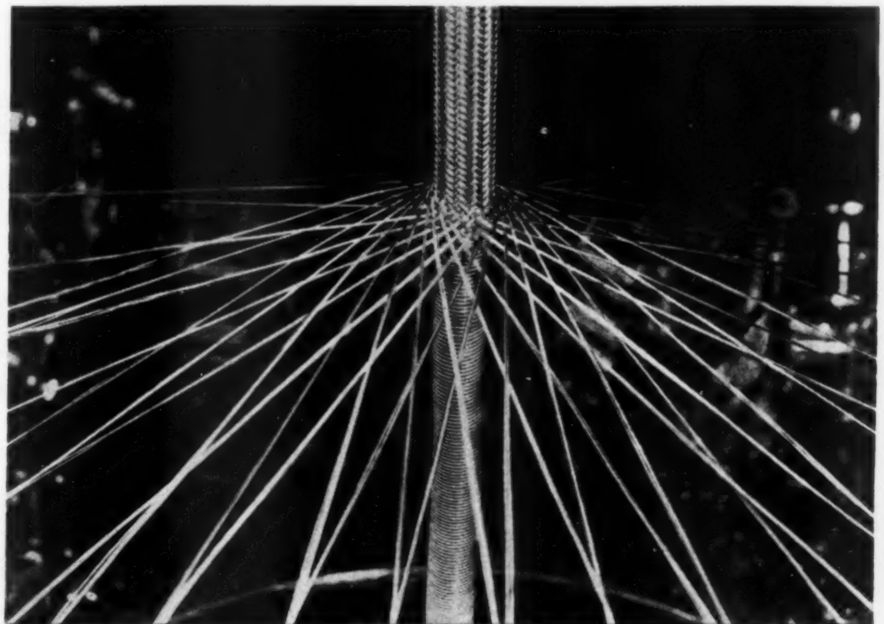
A sandwich of aluminum sheets and honeycomb is used for the dive brake door used on North American F-86 Sabrejets. Sandwich construction is lighter than that using metal reinforcing ribs, but this weight saving would be useless without an adhesive strong enough to withstand the stresses of high speed flight. A black, sticky mass called Epon is the adhesive used and, though somewhat difficult to apply, forms a sure bond when correctly applied.

Honeycomb is received in large tightly compressed sheets from which narrow strips of predetermined size are cut. Metal bars are fastened to each end of the strip, and it is then stretched to give it cellular form. The stretched honeycomb is marked with templates and then cut to shape on a bandsaw. A special fixture holds the honeycomb cut to correct outline while it is bandsawed to correct contour so it will conform to the curved outer surface of the dive brake door.

Cheesecloth is thoroughly impregnated with Epon, which must be applied very evenly. The impregnated cheesecloth is then applied to

the interior of the outer skin and prepared honeycomb placed in position on the cheesecloth. Skin and honeycomb are then tightly wrapped to a vacuum table which holds them tightly together during baking to cure the adhesive. Inner skin is then applied to the assembly by a similar process. Riveting of edges completes the process and the door is ready for shipping.

(1) Sheet honeycomb is bandsawed into strips of correct size
(2) Stretched honeycomb is marked with templates and a sharp knife for cutting. Tape outlines the marks
(3) Cheesecloth is thoroughly impregnated with Epon, which must be applied very evenly
(4) Impregnated cheese cloth is applied to skin and honeycomb sections put in position
(5) Assembly is held together by vacuum during baking
(6) Completed dive brake door



MAYPOLE?: This "maypole" is actually metal braid being applied to the inner core of Titeflex metal hose. Braiding machine may have from 16 to 48 braid carriers, each carrier holding from one to eight strands

Munitions Board Urges Avoiding Critical Materials

In a study recently completed by the Munitions Board, Department of Defense, as a guide for selection of materials for military engineering and design, a group of materials, listed in Column A, have been designated as strategic materials in short supply. It is advised that uses for these materials be reduced, with substitutes used whenever possible. In Column B are materials available in necessary quantities to meet emergency requirements.

A
Materials in Short Supply in Time of Total War

Nickel
Cobalt
Tantalum
Columbium
Beryllium
Tungsten
Copper
Cadmium

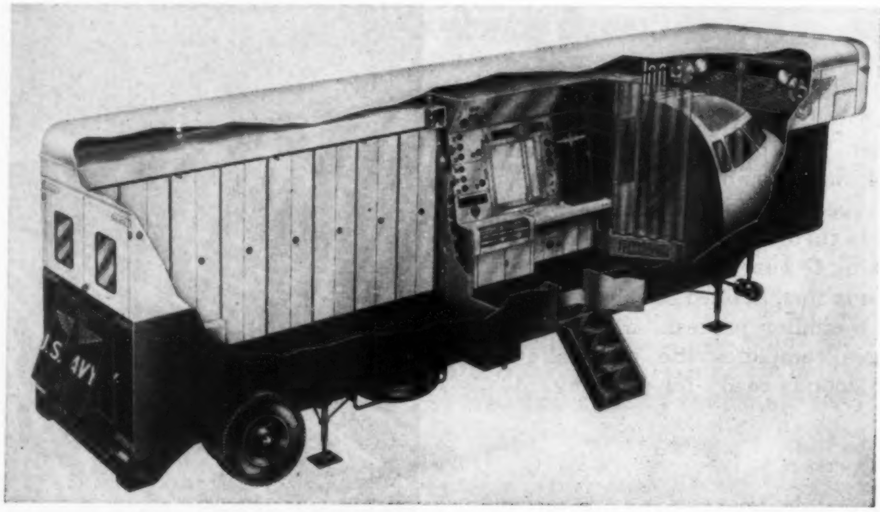
B
Materials Readily Available

Molybdenum	Boron
Titanium	Silicon
Chromium	Manganese
Manganese	Antimony
Zinc	Lead
Aluminum	Magnesium
Vanadium	

At present, moderate substitution of molybdenum for tungsten is desirable, since the supply of molybdenum would become critical if general substitution occurred. Particularly urgent is the evaluation of uses for nickel. Development of alternate low alloy steels is recommended. Also advised is the use of oxide, ceramic, nonmetallic paint or plastic coating instead of metals now used as protective coatings.

Awards in the 1952-1953 Heli-Coil Engineering Student Design Award Program were made as follows: first award of $1000 to James D. Dunfee of the Drexel Institute of Technology; second award of $500 to Joseph F. Klipp, a student at New York University; and because of a tie, duplicate third awards of $250 to Warren C. Bross of the Newark College of Engineering and Robert W. Bradspies, a student at the New York University College of Engineering. In addition, merit awards of $100 were made to Midshipman Ronald E. Adler, 1/c USN, of the U. S. Naval Academy and R. J. Murphy of Stevens Institute of Technology.

Navy Pilots Fly High Without Leaving Ground



Cutaway sketch shows layout of cockpit section, instructor's console and analog computer racks. Entry through the instructor's station isolates pilots from unrealistic distraction

Future pilots of the Navy's P2V-5 long range flying arsenal are receiving flight instruction in the new P2V-5 Flightronic simulator, an airplane trainer housed in a 42 foot mobile trailer.

Illusion of flight in these simulators is complete to clouds, lightning, structural vibration and propellor "chirp" when the engines are idling. Designed and built for the Special Devices Center, Office of Naval Research, by Engineering and Research Corp., this new trainer provides an economical and safe method of training pilots.

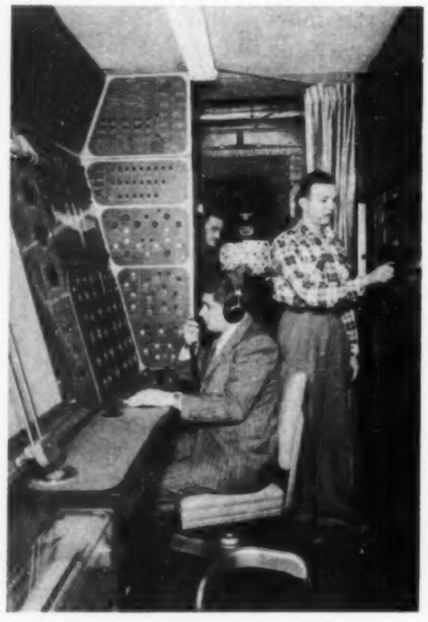
At the chart projection system, the instructor selects any of six chart scales by rotating a disc of color slides accurately aligned with the plotting board. Different areas are covered by changing the disc in the projector

Crash landings and other emergency procedures (caused by switches in the instructor's compartment) enable the pilots to become thoroughly familiar with failures liable to occur in actual flight. Radio communications and aids to navigation are also incorporated in the trainer.

Basically, an electronic computer in which all the forces acting on the airplane in flight (including the pilot's control forces) are considered as items in a number of equations, the trainer provides a type of simulation far more complete than types formerly used.

A new feature in the design of this simulator is the possibility of coupling the device with a P2V-5 tactics trainer, now nearing completion. This will be the first time the entire combat crew of a large military airplane will be able to perform operation and tactics training together in a flight simulator. This combination will probably be the largest and most comprehensive synthetic device ever built.

From the instructor's console, left, the entire flight procedure of the pilot is monitored. At the right, the engineer can adjust one of the three radio station simulators to simulate any type of radio aids transmission, which can be controlled to apparently originate from any desired position on the plotting board map



New Viscosity Standard

National Bureau of Standards has adopted the value of 0.01002-poise for the absolute viscosity of water at 20 deg C as the primary standard for the calibration of standard viscosity samples and viscometers. The American Society for Testing Materials, the National Physical Laboratory in England and the Physikalisch-Technischen Bundesanstalt in Germany have also indicated that they will adopt this value as of July 1, 1953.

The value of 0.01005-poise for the absolute viscosity of water had been widely used as the primary reference standard. Use of the new value of 0.01002 will result in a reduction of 0.3 per cent in the measured values of viscosity and will make viscosities reported in absolute units correspondingly more accurate. Previously published data based on 0.01005-poise may be adjusted to the new standard by reducing the published values by 0.3 per cent.

LOW COMPRESSION-SET RUBBER: A new low compression-set (8 per cent) silicone rubber, recommended particularly for applications where parts such as seals and gaskets must remain effective under sustained pressure, has been developed by the General Electric Co. The two molded disks were both compressed 50 per cent and baked for 22 hours at 300 F. The general purpose material recovered only 10 per cent of its original height height while the new SE-360 recovered 92 per cent

Adapt Synthetics To Electrical Insulation

New in the field of electrical insulation, Dacron, a compound of ethylene glycol and terephthalic acid previously used for textiles only, is undergoing wide commercial evaluation as a cover for dry transformer conductors. This Du Pont polyester displays high resistance to degradation by heat, low moisture absorption, chemical resistance, great stretch resistance and strength. Dacron, which can be made tougher than asbestos, is said to be easy to apply as an insulating cover.

Orlon acrylic fiber, another textile material, is also being studied as an insulating material. Yarn of this synthetic in 75, 100 and 150-denier weights is being used mainly as insulation for magnet wire. Orlon winds readily and resists weathering and outdoor exposure to acid gases in the air.

Nylon, a relative veteran insulating material, is used as a wrapping on wire in magnets and in coils and transformers in electronic devices such as radar, sonar and other computers, direction finders and calculators.

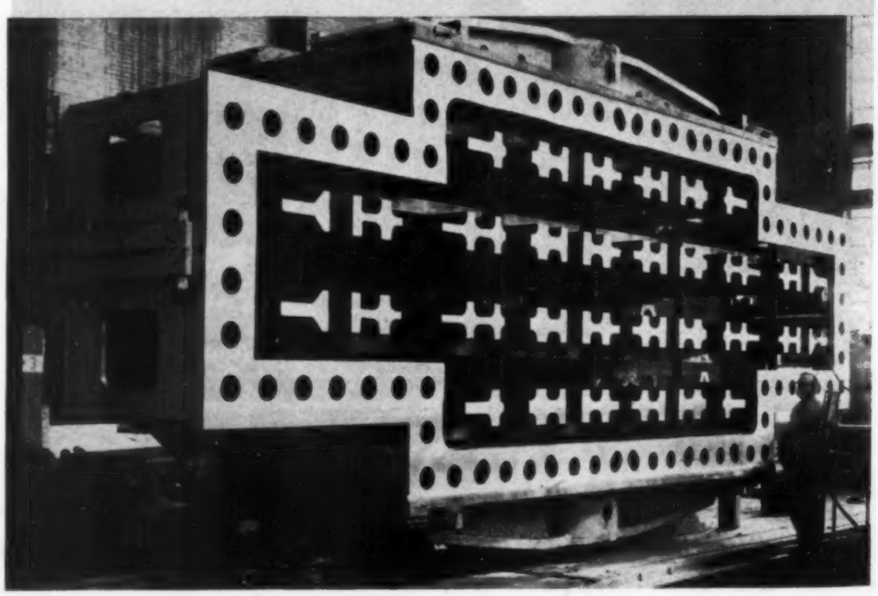
Corrosion Measurements Made with Interferometer

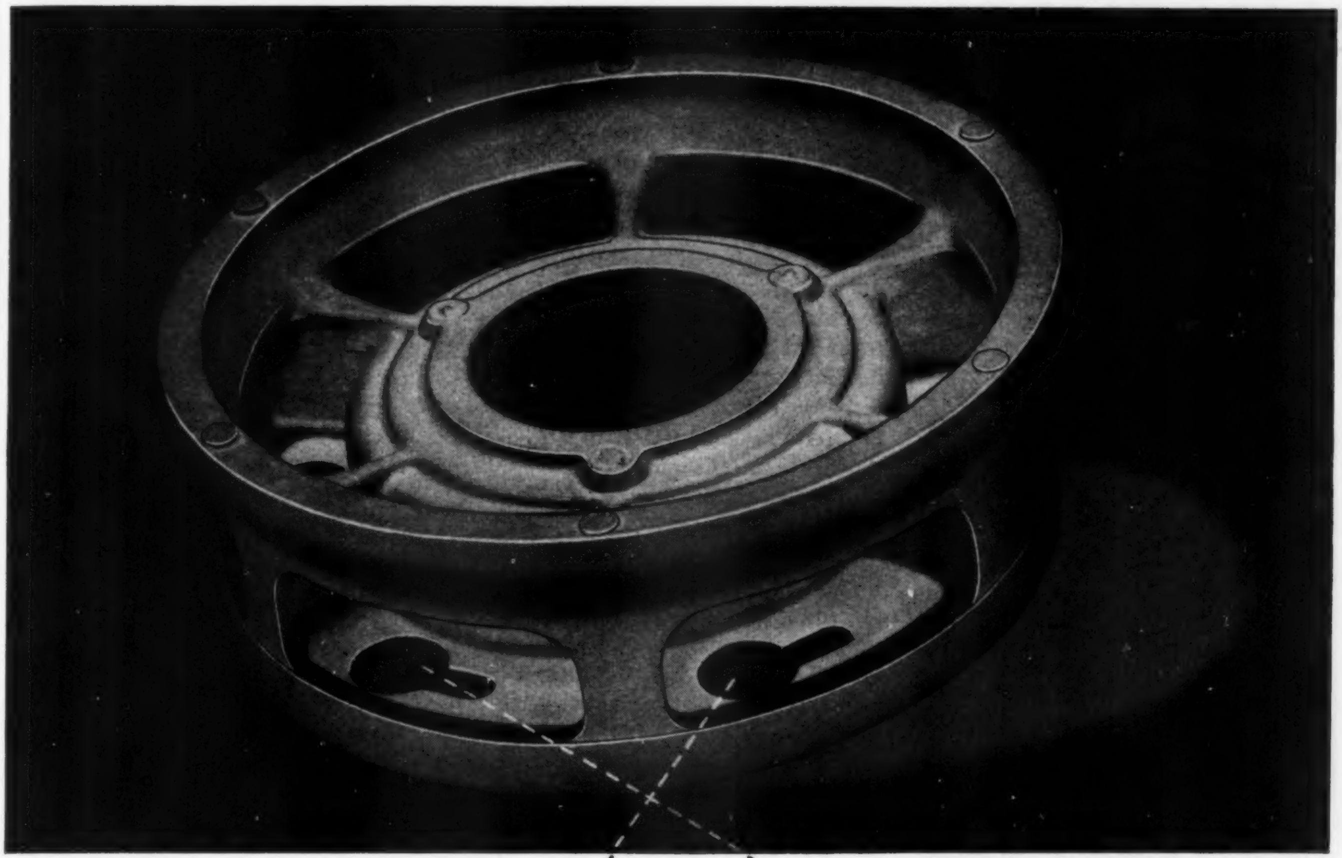
Studies of the corrosion resistance of such materials as optical glass, porcelain enamel, quartz and other natural and artificial silicates, and various metals are being made at the National Bureau of Standards by means of interferometric measurements. Comparison and prediction of the chemical durability of various materials is believed possible with this method. Corrosion to a depth of as little as 1 or 2 ten-millionths of an inch can be detected on optically flat specimens.

Specimens are ground and polished to optical flatness and then immersed to half their depth in the desired corrosive solution. After a specified time they are withdrawn, rinsed and dried and covered with an optically flat piece of quartz. Using a conventional interferometric viewing apparatus of the Pulfrich type, and with an unfiltered helium lamp for illumination, displacement of the optical interference fringes at the level-of-solution line of the specimen is observed. As viewed through the eye-

Interferometer in use at National Bureau of Standards measures amount of corrosion of an optically flat specimen

MASSIVE PRESS SLIDE: This unusual assembly is the draw slide of an 8000-ton triple-action hydraulic press for stamping out integrally stiffened air frame sections at Lockheed Corp. The slide is approximately 31 feet long, 14 feet wide and 11½ feet high. Designed by Birdsboro Steel Foundry and Machine Co., the assembly was machined and fabricated by Bethlehem Steel Co.





*FROZEN MERCURY INVESTMENT CASTING

MERCAST* Buttonholes

BUTTON A GENERATOR on or off of an airplane? Westinghouse does! And at a reported time saving of up to 8 man-hours per plane!

The ingenious new Westinghouse generator mounting bracket employs "buttonholes" that permit the assembly to be quickly slipped on or off over headed studs on the airframe. For practical production of this unit, a one-piece precision casting to close dimensional and surface tolerances is the economical answer. This casting, however, is much larger than can be produced by ordinary investment casting processes.

Westinghouse brought the problem to Alloy Precision Castings Company. Engineers and metallurgists at Alloy developed the illustrated 6 lb., 6½ inch diameter mercasting in 410 stainless. Critical tolerances are held to ±.003"/inch. Very little finish machining is required and the "buttonholes" are used as cast. Complete inspection by Alloy assures customers top quality and accuracy.

Mercasting made the Westinghouse generator mount idea a reality. Send details about your "impossible" or "too costly" jobs today! Learn how Alloy Precision can save you time and money with designs for mercasting.

ALLOY PRECISION CASTINGS COMPANY, DEPT. C-4
45th and Hamilton, Cleveland 14, Ohio

Please RUSH Bulletin 706 describing the Mercasting Process.

NAME ____________

TITLE ____________

COMPANY ____________

ADDRESS ____________

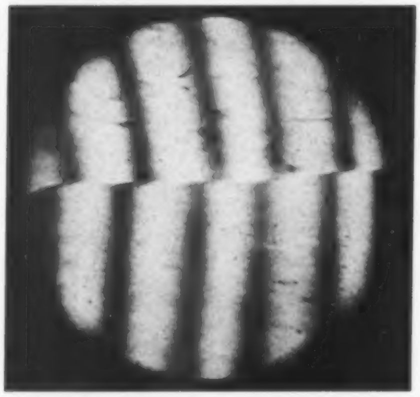
CITY ____________

ZONE ______ STATE ____________

ALLOY PRECISION Castings Company

EAST 45th ST. AND HAMILTON AVE. • CLEVELAND 14, OHIO

piece of the instrument, the interference fringes are vertical parallel lines. If the uniform flatness of the specimen being studied has not been impaired by the corrosive solution, these fringes are straight and continuous. However, if the surface has been attacked where immersed, each of the vertical



Typical fringe pattern as seen through the eyepiece of the interferometer. Swelling of the lower half of the specimen, which was immersed in a corrosive solution, has caused displacement of the fringes to the left

fringes is shifted laterally at the point where it crosses the line where thickness changes. If as sometimes happens, the solution has caused swelling of the specimen, the lateral shift is in the opposite direction. A lateral shift equal to the space between two adjacent fringes corresponds to a dimensional change in the specimen amounting to one-half wavelength of light, or approximately 0.3 micron. With careful use of the interferometer, changes in specimen thickness of as little as 0.003-micron can be detected.

A 51 year old high school in Rockford, Ill. is being converted to house both engineering and production facilities of the John S. Barnes Corp. Extensive remodeling will add a new modernistic entrance, passenger and freight elevators, truck and rail loading docks and a saw-tooth roof for improved illumination. Production at the plant will be for the automotive and machine tool industries.

EMC Censures Military Manpower Control

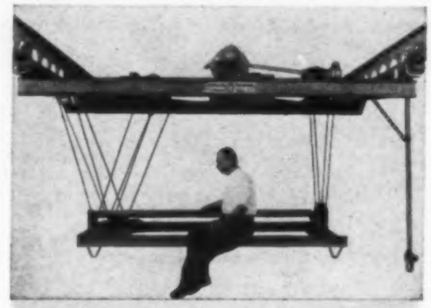
Twenty-five per cent of the engineers and scientists now in productive defense, development and research could be placed in uniform by a snap of the Pentagon's official finger, according to a recent statement by the Engineering Manpower Commission of the Engineers Joint Council.

This situation exists due to a compulsory reserve, created by Public Law 51, into which military personnel including Selective Service Inductees and ROTC graduates go upon completion of active service. This reserve pool, which is expanding every day, will soon exceed 10 million men and includes an increasing proportion of the engineers and scientists vital to our defense production economy. Recall of these reservists rests solely with the Department of Defense.

To prevent such an occurrence, the Engineering Manpower Commission and ten of the nation's largest professional and trade associations are backing legislation known as the Flanders-Johnson Bill which, they say, can save the nation from a major catastrophe.

"There! Now who says I can't even design one of these!

West Coast Electrical Mfg. Corp. recently moved into new quarters at 233 W. 116th Place, Los Angeles 61, Calif. All business is now being conducted from the new address. Adjacent property will permit future expansion.



STABILIZED CRANE: Load swing is eliminated in a new Cleveland Tramrail crane which finds use in tank-dipping operations, in various assembly jobs, and for supporting traveling X-ray equipment used in making studies of long objects. Hoisting ropes form a triangular suspension which permits holding a load rigidly in place and eliminates longitudinal, lateral and rotational sway

Consolidation of Timken-Detroit Axle Co. and Standard Steel Spring Co. to form Rockwell Spring and Axle Co. is expected early this fall. The new company, with 21 plants employing 19,000, will be a supplier to nearly every passenger car, truck, trailer, farm equipment, road building equipment and material handling equipment manufacturer.

Packing Manual

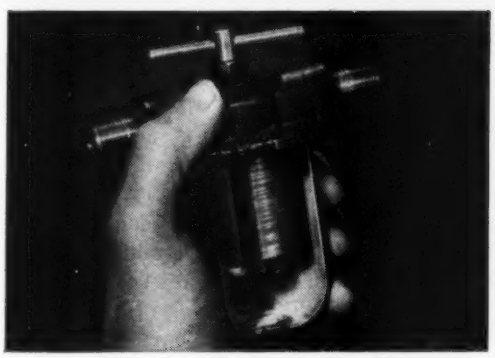
A recently published 60-page manual and catalog contains application data on leather and synthetic rubber packings as well as O-rings. Complete information on types of packings meeting ASTM, AMS, JIC and government standards is also included. Factors to be considered in choosing the correct material for a particular type of service are fully discussed in the publication, designated Catalog and Manual 201 by the publishers, Graton and Knight Co. of Worcester, Mass. and International Packings Co. of Bristol, N. H. A copy may be obtained from Graton & Knight at 356 Franklin St., Worcester, Mass.

Here's how this strainer "combs" itself clean!

This simple chalk test shows how AUTO-KLEAN's unique built-in comb construction cleans the strainer without costly interruption of flow

ORDINARY BLACKBOARD CHALK leaves heavy deposit of chalk particles on and between discs of Cuno AUTO-KLEAN strainer.

TURNING HANDLE ONE REVOLUTION moves strainer element through comb blades, removing all traces of chalk from between discs. Cuno's exclusive combing operation cleans *thoroughly*—without stopping flow.



FILTERING AREA IS COMPLETELY CLEAN, restoring full initial capacity. All chalk particles and dirt fall to bottom of housing where they can be drained periodically.

- AUTO-KLEAN's permanent metal filter element is available in steel, brass, or stainless steel for long, trouble-free service.
- AUTO-KLEAN is adaptable to any fluid-flow system.
- From acids to tar . . . if you can pump it, Cuno can filter it. Capacities range from one gallon per hour to 15,000 gallons per minute.

CUNO **ENGINEERED FILTRATION**

Removes More Sizes of Solids From More Kinds of Fluids

AUTO-KLEAN (disc-type) • MICRO-KLEAN (fibre cartridge) • FLO-KLEAN (wire-wound)

SINGLE-STRAINER PROTECTION FOR LUBE OILS AND PROCESS FLUIDS

Continuously cleanable AUTO-KLEAN eliminates need for stand-by strainers

AUTO-KLEAN's compact construction gives you full-flow operation in space which would limit ordinary filters to by-pass service. The low pressure drop of AUTO-KLEAN strainers permits this *full-flow* service on gravity, low pressure, high pressure, or suction lines—with no loss in operating efficiency.

Cuno's exclusive "comb-clean" action provides complete cleaning of the filter element—without stopping fluid flow. Thus, there's no need for a stand-by strainer.

The low maintenance costs of the AUTO-KLEAN save you money, for there are no cartridges to change. An occasional rotation of the handle does the cleaning job (most units can be equipped with motor-drives for continuous cleaning).

Fixed-space metal discs in this modern strainer positively remove all solids larger than the specified disc spacing—from .0035" up to .062".

For permanent, positive fluid protection, install compact Cuno AUTO-KLEAN—the precision-built strainer. Send the coupon for free bulletin.

A.3.3

Cuno Engineering Corporation
Dept. 203A South Vine Street, Meriden, Conn.
Please send me bulletin on Cuno AUTO-KLEAN
for ....
(fluid)

Name..............................

Company............................

Address............................

City.............Zone....State........

Helicopter Has Four-Bladed Rotor

The first test flight of a new evacuation ambulance helicopter, the Doman YH-31, was completed recently. Unique in that it has a four-bladed single rotor, the helicopter also features a compact rotor unit, large capacity, 7-foot wide doors, tail rotor above normal head height and constant pressure of oil flow in the rotor system for all moving parts.

An innovation is the use of a fluid coupling to initially engage the rotor system to the enginc and an automatic lockout to provide a positive drive system between the engine and rotor in all flight conditions. Compound exhaust ejectors draw cooling air over the engine cylinders, eliminating the need of a cooling fan

Sister ship of the YH-31, Doman's LZ-5 has been converted to a twin-engine model with over 1 ton useful load capacity. This ship will make possible the expansion of the commercial helicopter field by providing inter-airport and city-to-airport transportation, according to company president Glidden S. Gilman. Both the YH-31 and the LZ-5 feature the Doman four-bladed lifting rotor, called the world's only successful hingeless rotor.

Vacuum coating apparatus for depositing rhodium and other metals

Rhodium Coating Service Available

Conductive and nonconductive materials can be coated with rhodium by using the vacuum technique, according to Serfass Corp. Rhodium is being applied on a commercial basis upon glass and other nonconductive substances to meet specifications of transmittency and hardness. Coating thicknesses can be applied to give controlled reflectivity or transmittance within 5 per cent.

Rhodium has high resistance to chemical attack, and because of its high reflectivity, is used for mirrors and reflecting surfaces which must withstand corrosive chemical atmospheres. It has been deposited on glass, microscope slides, mirrors for optical apparatus, reflectors for telescopes, glass go and no-go gages and glass bearing surfaces. When deposited on glass and baked, rhodium coatings become extremely hard and therefore provide thin, hard surfaces which resist wear and friction.

Hard Vinyl Plastics Welded or Machined

Two recent developments promise to widen the already extensive field for nonplasticized hard vinyl plastics. Welding rods of triangular cross-section, 0.20 and 0.28-inch on a side, expedite the welding of large assemblies such as chemical sinks and corrosion resistant fans; and solid bars up to 2.50 inches in diameter and of uniform density permit the machining of screws, nuts, bolts, gears and valves.

Welding of plastics was formerly a costly and time consuming operation, as in any but the narrowest seams, filler material was required and only repeated passes produced satisfactory seams. The new Lucoflex triangular welding rods are said to produce tight seams without filler materials and with a single pass. Economies of 80 per cent are visualized in time alone; materials costs are expected to be reduced by 50 per cent, according to Lucoflex engineers.

Up to now, large rigid assemblies constructed entirely of corrosion resistant plastics were rare because large bars are often spongy, making machining difficult at best, and producing rejects. The alternative was to use metallic supports at the sacrifice of corrosion resistance. The new Lucoflex bars, 2.00 and 2.50 inches in diameter, are claimed to have uniform density, mechanical properties and good chemical resistance.

Liquid sodium is being pumped at temperatures of 580 to 1000 F in an experimental installation at the Knolls Atomic Lab, operated by GE for the AEC. Purpose of the installation is to study characteristics of electromagnetic pumps and flowmeters which are now commercially available.

Contact KAYDON of Muskegon

FOR ALL TYPES OF BALL AND ROLLER BEARINGS: 4" BORE TO 120" OUTSIDE DIAMETER

Special KAYDON two-row Angular Contact Ball Bearings, 12.000" x 13.440" x 1.375" (THIN SECTION)

These KAYDON-bearinged BELL Anti-Submarine Helicopters won't be outsmarted!

NAVY'S latest "bad news" for unwelcome submarines are the HSL-1 Bell Helicopters . . . most powerful tandem-rotor 'copters known . . . big but compact, with high speed, long range and all-around performance "beyond expectations".

Special thin-section **KAYDON** two-row angular contact ball bearings, used on the swash plates, contribute much to the compactness, speed, controlability and stability of these modern defenders of America. • **KAYDON** cooperates with many designers and technicians in terms of precision bearings that improve aircraft, automotive, military and industrial equipment . . . to help make many products better, faster, more profitably.

On units you *make* to *sell*, or *buy* to *use*, specify **KAYDON** bearings. Contact **KAYDON** of Muskegon.

THE KAYDON ENGINEERING CORP.

MUSKEGON • MICHIGAN

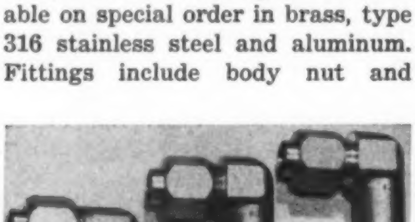
KAYDON Types of Standard and Special Bearings: Spherical Roller • Taper Roller • Ball Radial • Ball Thrust • Roller Radial • Roller Thrust • Bi-Angular Bearings

PRECISION BALL AND ROLLER BEARINGS

NEW PARTS

Elbows

Parker Triple-lok fitting line has been expanded to include long and extra long male elbows. Addition of these fittings affords a greater flexibility in planning tube circuits. Shorter, standard shaped elbows may be nested within the longer elbows, permitting close, parallel lines. Standard stock is steel; however, the two new shapes are available on special order in brass, type 316 stainless steel and aluminum. Fittings include body nut and



sleeve. Shown left to right, are the standard C BTX, the long CC BTX and the extra long CCC BTX male elbows. Made by **Parker Appliance Co.**, 17325 Euclid Ave., Cleveland, O.

For more data circle MD-34, page 203

Stainless Steel

Alloy V2B combines high hardness, nongalling characteristics and corrosion resistance. It is a hardenable 18-8 type of stainless steel, containing copper, molybdenum, silicon and a very small amount of beryllium. Readily machinable in the quench-annealed state, it may be hardened by a low-temperature heat treatment, producing no distortion and only a light heat tinting discoloration, which may be removed. In the annealed condition, the material is easily welded using special V2B welding rods. V2B does not over-age at elevated temperatures and may therefore be used in steam applications and at temperatures up to 1400 F. Produced in both cast and wrought form. Made by **Cooper Alloy Foundry Co.**, 100 Bloy St., Hillside, N. J.

For more data circle MD-35, page 203

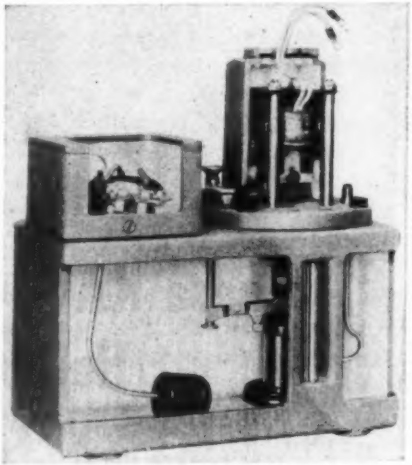
Vibration-Noise Isolator

The Type 915 Barrymount is designed to provide isolation at frequencies above approximately 20 cps, for equipment applying static loads of 15 to 200 lb per unit isolator. When loaded within its recommended range, it will provide from 70 to 90 per cent isolation at 30 cps (1800 rpm), and more at higher frequencies. Made by the **Barry Corp.**, 700 Pleasant St., Watertown 72, Mass.

For more data circle MD-36, page 203

Automatic Lubricator

Control flexibility is one of the features of the type E solenoid-operated lubricator which can be applied to almost all machines and mounted in any location. It requires no connection to a moving element of the machine. It can be actuated by trip switches, existing control circuits, electric timer con-



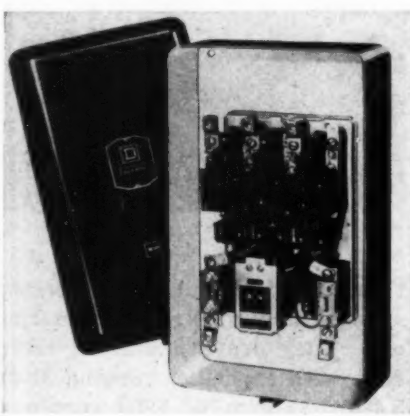
trols or pushbuttons. It will deliver from 6 to 24 drops of oil per stroke, and maximum frequency of operation recommended is four strokes a minute. Temperature rise is within safe limits with the solenoid cover in place, and the unit is available with either a 3 or a 6-pint capacity reservoir. The lubricator can be used on machine tools, metalworking machinery, packaging, bottling, printing, textile, woodworking and special and general industrial machines. Available as

an extra is a float-switch control that actuates a visible or audible alarm when the oil supply is low, and that will shut off the machine if the supply is not replenished. Made by **Bijur Lubricating Corp.**, Rochelle Park, N. J.

For more data circle MD-37, page 203

Vertical Action Starter

Featuring front mounted parts for accessibility and easy maintenance, new NEMA size 4 vertical action starter has maximum polyphase ratings of 50 hp, 220v and 100 hp, 440 to 550 v. Overall dimensions are 9¼-in. wide, 14¾-in. high and 7 3/16-in. deep. Double-break silver contacts do not require

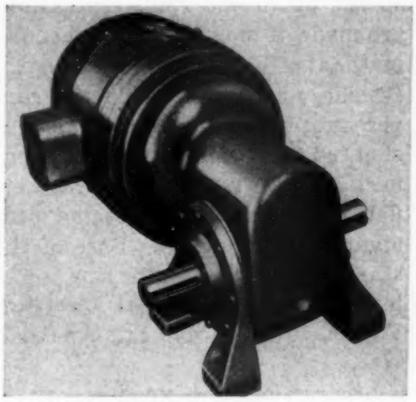


filing or dressing and eliminate flexible jumpers and pivots. The armature has straight line motion, guided top and bottom. As many as eight electrical interlocks can be added to the starter, four normally open and four normally closed. Made by **Square D Co.**, 4041 North Richards St., Milwaukee 12, Wis.

For more data circle MD-38, page 203

Right-Angle Motoreducer

This unit converts the high speed of a motor into low-speed, high-torque output. Right-angle, worm gear design offers economy and versatility in mounting and load hook-up. The unit permits the simultaneous use of as many as 3 output shafts from the same motor. Drive motor can be either standard squirrel-cage, slip ring, multispeed or fluid shaft in either dripproof, splashproof or totally enclosed de-



signs. Motoreducer is available in ½, ¾, 1 and 1½ hp ratings with speed reductions to as low as 18, 24, 36 and 58 rpm, respectively. Foot or ring mounting is optional. **Reuland Electric Co.**, Alhambra, Calif.

For more data circle MD-39, page 203

Barrel Nut

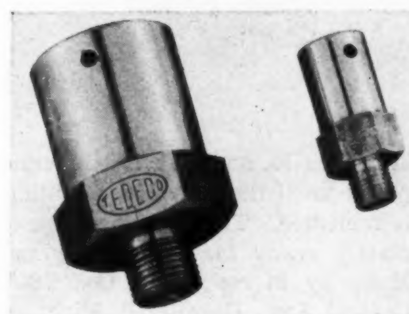
Self-locking Flo-Nutt can be used wherever conventional barrel nuts are employed. It eliminates the need for milling of large recesses for wrench clearance and is self-aligning. Where a number of matched drilled holes in one pat-

tern are required, this nut can be set up with a variance of 1/64-in. in any direction. This variance is possible because the self-aligning or "floating" nut can shift within the barrel. Body of the nut is an anodized aluminum alloy with a cadmium plated steel retainer. Available in a range of bolt sizes from ¼-28 to 1½-12 with tensile strength of 5760 to 272,000 lbs. Made by **A. A. Metal Products Inc.**, 6200 South Central Ave., Los Angeles 1, Calif.

For more data circle MD-40, page 203

Breather Vent

Originally designed for gear cases on aircraft, this vent has internal baffles to prevent oil spillage, contains no corrosive material and conforms to Naval and USAF aircraft specifications. It relieves internal pressure and permits filtered



air to enter the gear case. Installation requires a tapped hole near the top of the gear case housing. Available in several standard or special sizes from **Technical Development Co.**, 6740-42 Baltimore Ave., Fernwood, Pa.

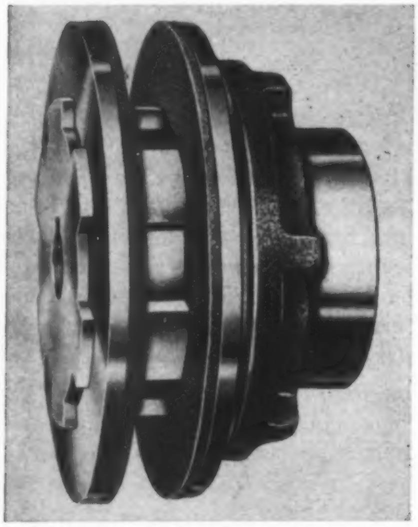
For more data circle MD-41, page 203

Torque-Limiting Clutches

Slip type friction clutches for use as overload protective devices in power drives whose torque capacity does not exceed 260 lb-ft are designed as an alternative to shear pin methods. Three models are made with diameters of 4½, 5½ and 6½-in. and ratings of 55, 121.5 and 260 lb-ft respectively. Units are particularly adapted to roller chain drives and are designed to be used in connection with standard Morse type A plain plate roller chain sprockets. Other types of

rotating members such as gears, pulleys and sheaves can also be used.

Clutches slip at the preselected overload torque setting, which can be adjusted within individual mod-

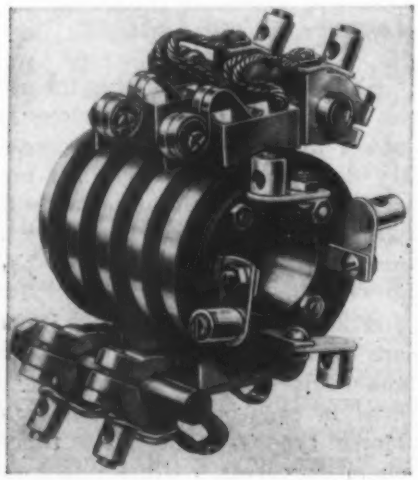


el capacities, and will not transmit power until the overload condition is remedied. They are then immediately ready for further service. Made by **Morse Chain Co.,** 7601 Central Ave., Detroit 10, Mich.

For more data circle MD-42, page 203

Collector Rings

Used for conducting current from a stationary to a rotating part, these collector rings require a minimum of space. Sizes range from 1 to 16 conductors, with conductor

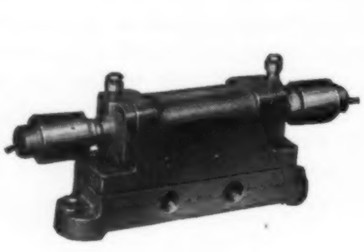


ratings from 35 amp at 600 v to 75 amp at 600 v. Industrial all-purpose collector rings are rated for speeds up to 900 rpm. Insulators are made of moistureproof phenolic material of high dielectric strength. Brushes are of high conductive copper graphite, and each brush is shunted for absolute electrical connection. A free-acting adjustable flat helical spring maintains constant brush pressure. Bore of the standard model collector ring is 1½ in. Made by **Industrial Electrical Works,** 1501 Chicago St., Omaha 2, Neb.

For more data circle MD-43, page 203

Pilot-Operated Valves

Model V aluminum four-way slide valve can be operated by cam, hand or foot-control poppets, or by solenoid pilots. This type R (reciprocating) master valve unit embodies a slide valve linked to a balanced piston with line pressure

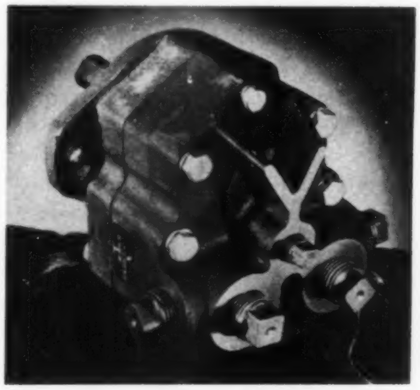


maintained on the inside as well as on both ends of this piston. Exhausting air from either end of the piston by means of two-way bleeder valves causes pressure on opposite end to shift the valve. Valve position is maintained until the air from the opposite end is bled. Type AR (air return) master valve unit also embodies a slide valve, in this case linked to an unbalanced piston; inside of the piston is under constant pressure. Piston is shifted by admitting compressed air to the large end through the energization of a three-way, normally closed solenoid pilot valve. Piston returns to normal position when pilot valve is de-energized. All air passages are precision drilled. Type R is available in ¼, ⅜, ½ and 1-in. pipe sizes; type AR, in ¼, ⅜ and ½-in. pipe sizes. Available from Nopak Div., **Galland-Henning Mfg. Co.,** Milwaukee 46, Wis.

For more data circle MD-44, page 203

Hydraulic Pumps

HC Series gear type pumps are designed for use with agricultural implements, tractor loaders, industrial trucks, machine tools, agricul-



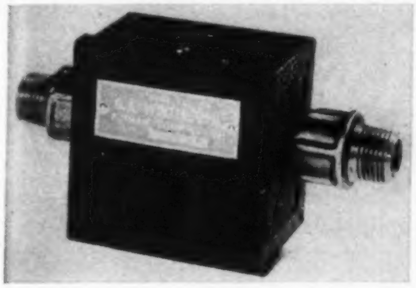
tural and industrial power steering systems and construction machinery. They are available in three sizes with capacities ranging from 5.4 to 10 gpm at 1800 rpm and pressure of 1000 psi, and will pump most liquids having a substantial oil base. The shaft design permits these pumps to be direct, gear, or belt driven. A wide selection of porting locations is available; end or side porting is standard. HC Series units may be had with an internal relief valve which permits regulation of pressures between 500 and 1200 psi, but cannot be used as a control or pressure regulating valve. Made by the **Webster Electric Co.,** Oil Hydraulics Div., 1900 Clark St., Racine, Wis.

For more data circle MD-45, page 203

Flow Switch

The magnetically coupled flow-indicating switch controls electrical circuits for signalling flow or non-flow of liquids through pipe lines.

It is suitable for use with water and other liquids not injurious to brass or copper. Flow is sensed by a movable vane within the flow tube. Variation from desired rate

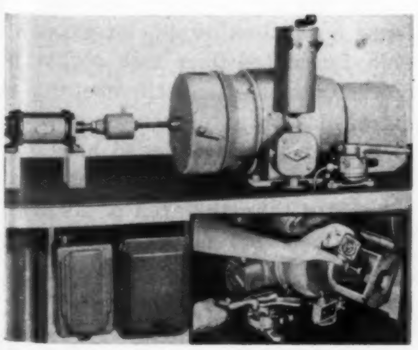


of flow causes a magnet affixed to the movable vane to actuate the contacts of a mercury magnetic tube outside the flow tube. Two models of the switch are available, one which allows a maximum flow of 2.5 gpm through the flow tube and can be adjusted for flow sensitivity of 0.1 to 0.6-gpm, and one with an adjustment range of 0.25 to 1 gpm and allowing a maximum flow of 4 gpm. This control unit can be used in lines of greater capacity by installation of a by-pass and regulating valve arrangement. Made by **Winterburn Mfg. Co.**, Putnam, Conn.

For more data circle MD-46, page 203

Magnetic Brake-Motor

This unit is said to be the first to use a magnetic brake in conjunction with a hollow-shaft motor. This permits the motor's shaft to extend entirely through the brake. The shaft passage is thereby kept open for rams or "punch out" rods,



which can pass entirely through the motor. Typical of set-ups using this unit is the one illustrated. It involves the lapping of three ports in a valve to a precise dimension. The operator first positions the valve in the fixture, locking it in place by activating the ram. The fixture, which is connected to the motor shaft, rotates with the valve when the motor is started. As the valve rotates, the operator laps the three ports simultaneously through the opening in the end of the fixture. The magnetic brake permits fast, automatic starting and stopping for the repeated insertion and removal of work. Additional uses for this self-contained unit include applications such as grinding, honing, spinning and turning. Made by **Reuland Electric Co.**, 3001 W. Mission Rd., Alhambra, Calif.

For more data circle MD-47, page 203

Hydraulic Cylinder

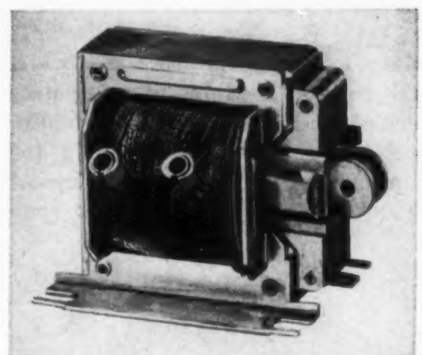
Key type cylinder has a long Meehanite rod bearing which provides a full one-to-one surface for extra wear. Chevron type packing is replaceable, and the whole bearing insert is held by a locking

snap ring at the rod end. Cast steel mounts are an integral unit combined with the head and rod caps of the cylinder. The head and mount combinations are easily disassembled by removing the internal locking key ring which unites the head and cylinder wall. Cylinders have heavy wall tubing, chrome-plated rods, rod scraper and milled wrench flats on rods. Cylinder heads are locked in place in a counterbored section of the tubing wall at the ends of the cylinder. Pipe ports can be oriented at random, since the key type construction permits 360-degree rotation of the ends. Sizes available are 1½ through 8-in. bores with any stroke length up to 18 ft; all standard mounts are available from **Carter Controls Inc.**, 2914 Bernice Rd., Lansing, Ill.

For more data circle MD-48, page 203

Solenoid

Replaceable coil of this 2000-size C-T solenoid can be changed in the field by removing two spring



clamps. Model 230 delivers 6.5 lb-in. per stroke, while No. 231 is rated 9.5 lb-in. per stroke. Both can be supplied for any voltage rating at 50 or 60 cycles. Field coil is moisture-proof and can be oil-proof, if desired. Double cotton braid lead wires are standard, with screw or solder type terminals available when specified. Frame and plunger are cadmium plated. Made by **Dormeyer Industries**, Dept. Z-1, 3418 N. Milwaukee Ave., Chicago 41, Ill.

For more data circle MD-49, page 203

Organic Metal Finish

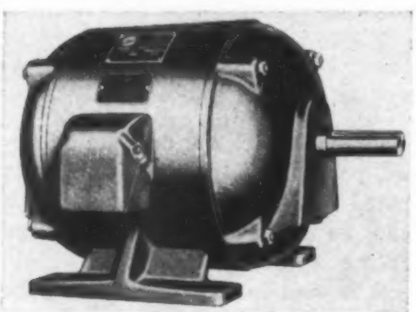
Nonfading and corrosion-resistant, a baked enamel organic finish can be applied to aluminum, brass, steel and zinc coils in widths from ⅛ to 30 in., 0.010 to 0.036-in. gage, in continuous lengths. The finish will not crack, chip or chalk even under abnormal use or severe weather conditions, and it is not

affected by forming, drawing, bending or fabricating of the metal. Typical applications include tubing, clips, screw caps, buttons, switch plates, battery tops, lighting fixtures, and electrical appliances. Surfaces can be plain or crimped. Available in standard colors of blue, brown, yellow, green, red, white, and black, as well as other special colors, from **Apollo Metal Works**, 66th Place and South Oak Park Ave., Chicago 38, Ill.

For more data circle MD-50, page 203

Induction Motors

Type RS, dripproof, continuous-duty motors in frame size NEMA 66 to 505 feature prelubricated, sealed ball bearings which require neither greasing nor cleaning for normal life. Rated at 1/3-hp at 1800 rpm to 150 hp at 3600 rpm,

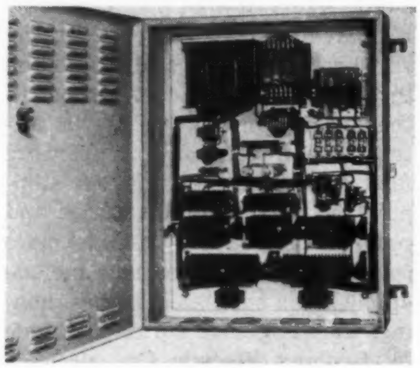


the motors are two or three-phase and operate at all standard frequencies and commercial voltages below 600 v. Available for regular, vertical, wall or ceiling mounting from the **Lima Electric Motor Co.**, Lima, O.

For more data circle MD-51, page 203

Adjustable-Speed Drive

This magnetic amplifier speed control contains no tubes and is easily mounted and installed. Speed remains nearly constant as the load on the motor changes—even at the low speed of 50 rpm the speed does not drop off when load is applied, but holds practically constant to loads of twice the rated motor torque. The drive operates properly with supply voltages ranging from 200 to 250 volts with no sacrifice in life; offers instant starting, high overload capacity, and proper operation over a wide ambient tem-

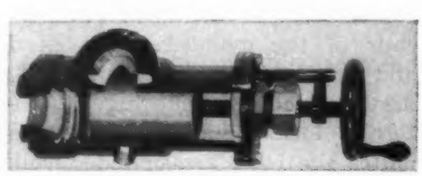


perature range. Control enclosure can be mounted out of the way of the driven machine, and the operator control station, containing the start-stop buttons and the speed control knob, can be mounted in any position convenient for the machine operator. Model E-1 includes a standard 230 v dc motor with a speed range from 49 to 2450 rpm and is rated at 3/4-hp at maximum speed. Developed by the **Franklin Control Corp.**, 1975 South Allis St., Milwaukee 7, Wis.

For more data circle MD-52, page 203

Flow Control Valve

Marketed under the tradename Anglomer, this valve controls flow of viscous liquids, such as paints,

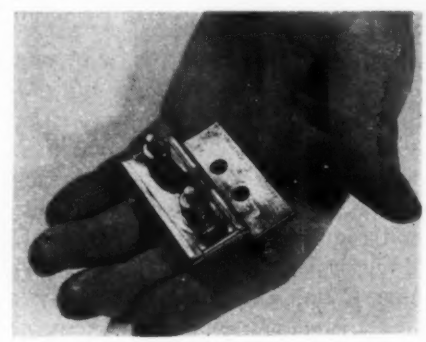


crude oil, or waste liquids containing quantities of dirt or sediment without its passages becoming clogged. Accurate control of varying amounts of flow is possible because the spiral passage in the valve body allows adjustment of internal friction on the liquid being controlled, pressure on the primary side remaining constant. Valve is available in 1¼, 2, and 2½-in. sizes. Manufactured by **Anglo American Mill Co., Inc.**, Owensboro, Ky.

For more data circle MD-53, page 203

Compact Spot Heater

Instruments exposed to sub-zero temperatures can be kept functioning by this small spot heater. The resistance element heater has a rating of 100 w—and operates on 28 v dc. Electrical connectors consist of double nut terminals with the negative terminal grounded to the case. Overall dimensions of the heater and the two closely



spaced mounting holes permit concentrated heating of flat surfaces as small as 1 x 3/8 inches. The unit has a maximum weight of 8 oz. Made by **Westinghouse Electric Corp.**, Box 2099, Pittsburgh 30, Pa.

For more data circle MD-54, page 203

Laminated Fiber Recharge

For use in full flow filtering of large quantities of oil, an expendable, laminated fiber disk recharge has four to ten times the useful life of cellulose, waste or redwood. It is classed as an extended area recharge because actual filtering area is greater than the area of its container. Disks remove foreign solids smaller than 2 microns, and absorption of these solids after 48 hours of draining at 70 F is 2½

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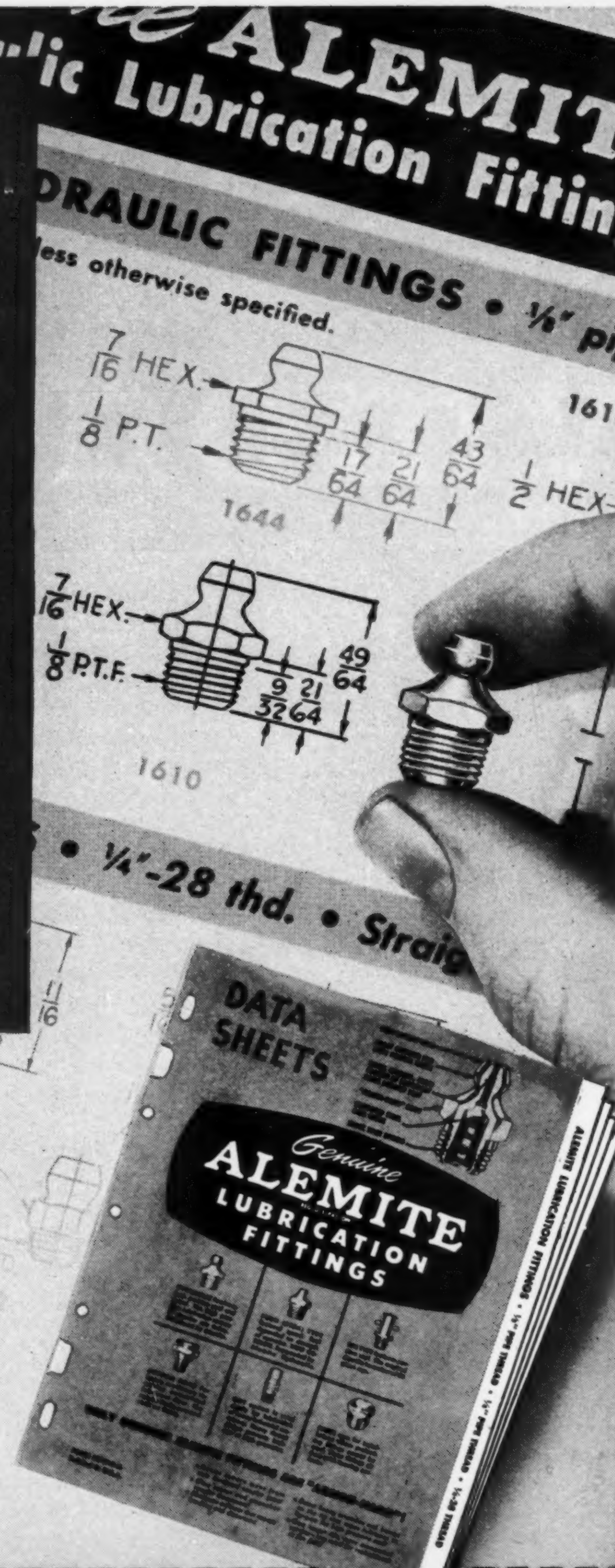

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times the weight of the recharge. Because no chemicals, bleaches or diatomaceous earths are used, the units will not remove vital additives from detergent oils. Developed by **William W. Nugent & Co. Inc.**, 410-12 North Hermitage Ave., Chicago, Ill.

For more data circle MD-55, page 203

Jacketed Pipe and Fittings

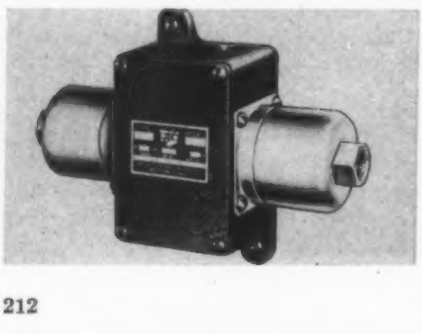
These welding fittings are available in 1 to 12-in. sizes in standard weight, extra strong schedule 160, and double extra strong. Complete

line consists of 90 and 45-degree elbows, tees and crosses. Made by **Red Jacket Co. Inc.**, 510 Investment Bldg., Pittsburgh 22, Pa.

For more data circle MD-56, page 203

Pressure-Vacuum Control

Improved type J6K sensitive differential pressure vacuum control has weatherproof and shock resistant features which make it suitable for exposed or heavy duty applications. It can be made watertight when specified. Switch controls the system differential up to 90 psi on applications requiring a

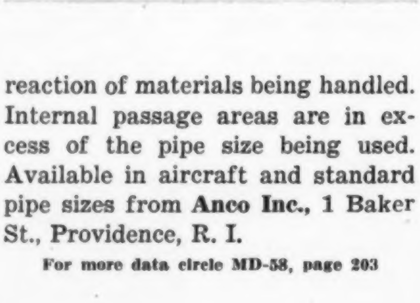


switch differential between 4 in. of water and 15 psi. Three switch actions are available, normally-open which closes on rising pressure or decreasing vacuum, normally-closed which opens on rising pressure or decreasing vacuum, and double-throw with no neutral position. Made by **United Electric Controls Co.**, 85 School St., Watertown 72, Mass.

For more data circle MD-57, page 203

Check Valve

This valve has a valve seat formed of two hardened, ground and lapped steel pieces. Consisting of seven parts in all, the valve has only one moving part. Used with accumulators, it prevents reversal of the flow of air, oil, water or chemicals, providing control lines with a minimum of pressure drop and positive sealing against return flow. Pressure is not lost through the valve, and cracking pressure is 5 lb or less at all pressures. Produced in steel, as well as forged aluminum for aircraft, the valve can be made to withstand chemical



reaction of materials being handled. Internal passage areas are in excess of the pipe size being used. Available in aircraft and standard pipe sizes from **Anco Inc.**, 1 Baker St., Providence, R. I.

For more data circle MD-58, page 203

Mechanical Coupling

By unusual application of rubber bushings, this Flexoid Silent-bloc mechanical coupling will allow parallel as well as angular misalignment up to 15 degrees with minimum shaft pressure on support bearings. It can be taken apart readily and eliminates lubrication,

noise and lost motion. Coupling uses two flanges fitted with encased rubber bushings, with four connecting pins floating freely inside inner sleeve of bushing elements. Angular deflection between plates is provided for by twisting action in the rubber. Couplings are offered in sizes 4B, 6B and 8B with bores of 1¼, 1⅝ and 2 7/16 in. and in ratings of 2, 4 and 8 hp per 100 rpm, respectively. Made by **Flexoid Coupling Co.**, 1545 East 23rd St., Cleveland 14, O.

For more data circle MD-59, page 203

Nut Retainer

Used with an external wrenching nut, this cadmium-plated steel retainer forms an anchor nut cap-

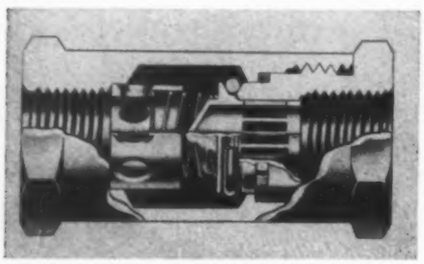
able of resisting 160,000 psi minimum tensile stress. Standard anchor nut rivet hole spacing provides

external wrenching nuts with thread sizes from $\frac{1}{4}$-28 through 1-14 from **Nutt-Shel Co. Inc.**, 811 Airway, Glendale 1, Calif.

For more data circle MD-60, page 203

Check Valve

For use with acids and other corrosives, these nonleaking stainless steel check valves employ Teflon O-rings and gaskets, utilizing the chemically inert properties of this material. Valves are supplied

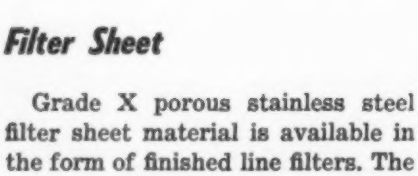


in various grades, depending on the required service, with male tube connections from $\frac{1}{4}$ to $1\frac{1}{2}$ in. or female pipe threads from $\frac{1}{8}$ to 1 in. Pressure rating is 3000 psi and maximum flow is 40 gpm. Tests conducted by the company indicate perfect performance and "dead-tight" sealing from temperatures of -65 F to 550 F. Made by **James-Pond-Clark**, 2181 E. Foothill Blvd., Pasadena 8, Calif.

For more data circle MD-61, page 203

Filter Sheet

Grade X porous stainless steel filter sheet material is available in the form of finished line filters. The



material has a tensile strength of approximately 25,000 psi and is more ductile than previous material of this type. Thickness tolerances are maintained to $\pm$0.002-in. Smooth surface sheet material is available from stock in 0.040-in. thickness, with a mean pore opening of 10 microns. Other thicknesses ranging from 0.020 to $\frac{1}{8}$-in. can be supplied by **Micro Metallic Corp.**, 35 Sea Cliff Ave., Glen Cove, N. Y.

For more data circle MD-62, page 203

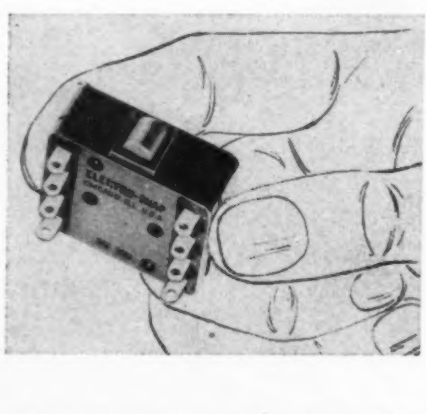
Phenolic Terminal Blocks

Designed for use in electronic and communications equipment, blocks are available with 40, 60, 80, or 100 pretinned, double-notched terminals securely fastened between phenolic strips. The terminal assembly is fastened to a base of the same material. Phenolic construction provides high dielectric strength. The metal mounting bar is tapped to facilitate installation. Made by **Lenkurt Electric Co.**, County Road, San Carlos, Calif.

For more data circle MD-63, page 203

Double Pole Switch

Developed for switching circuits having two different currents, phases or voltages, but which must be

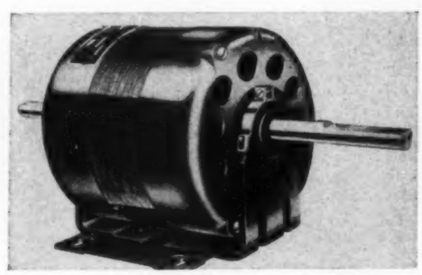


switched at the same time, this double-pole switch operates on the reaction principle. Because it can simultaneously break or reverse current flow through two windings of a three-phase motor, it can be used as a start-and-stop or limit switch on three-phase machinery. Made with eight terminals, the switch can operate up to four separate circuits at one snap. Case dimensions are $1\frac{1}{2}$ x $\frac{7}{8}$ x $\frac{1}{2}$-in. and rating is 15 amp at 125/250 v ac and 10 amp at 30 v dc. Simultaneous switching actions permit all four contacts to be wired for use as a single-pole switch on high voltage direct current circuits (110 v dc) up to 20 amp. Made by **Electro-Snap Switch & Mfg. Co.**, 4220 West Lake St., Chicago 24, Ill.

For more data circle MD-64, page 203

Fractional-Hp Motor

Model 5000 motor is designed for fan duty in air conditioners, window fans and similar applications. It is a six-pole shaded-pole unit with drawn steel housing and self-aligning bearings. Offered as a



two-speed or single-speed motor, it is available with or without rubber mounts and resilient base, and overload protection. Speeds of 1050 rpm with lower second speed of approximately 800 rpm are standard. Horsepower ratings offered are 1/15, 1/12, 1/10 and 1/8. Made by the **Electric Motor Corp.**, Div. of Howard Industries, Inc., Racine, Wis.

For more data circle MD-65, page 203

Fiber-Bonded Fabric

Dynel Windsor felt is a chemically resistant filter fabric for use in filter press or vacuum filter applications for efficient cake formation or clarity of filtrate. It is multiple-ply card-constructed for uni-

Facts about *HELI-COIL* inserts you should know

What they are
*Heli-Coil** screw thread inserts are precision formed coils of stainless steel or phosphor bronze wire. Wound into tapped holes, they form permanent, non-corrosive, strip-proof threads of astonishing strength. Available for National Coarse, National Fine and Unified threads, pipe threads and spark plug threads. They are made in all standard sizes and lengths for assemblies requiring Class 3, 3B, 2 or 2B fits.

What they are for
AS ORIGINAL COMPONENTS: *Heli-Coil* inserts are used to provide stronger, lighter fastenings, corrosion-proof, wear-proof threads in all assemblies.

FOR PRODUCTION SALVAGE: When conventional tapped holes are damaged in production, restore them on the line with *Heli-Coil* inserts. Get better-than-original strength with no increase in screw size and no tell-tale signs of rework.

FOR SPEEDY REPAIRS: When tapped threads wear, strip or corrode in service, renew them in minutes on location in shop or field with *Heli-Coil* inserts. No welding—no plugging—no secondary machining—no oversize screws.

How they work
Holes are drilled and tapped as you do for ordinary threads—then *Heli-Coil* inserts are wound into tapped holes by hand or power tools. Install in a few seconds, assure thread protection forever. Can be used in any metal wood or plastic.

No other method is so simple, effective and practical.

What they do for you
Heli-Coil inserts *save money* because they strengthen threads and make *fewer* smaller fastenings do the same holding job. They make *lighter* bosses and flanges practical and they *save weight* in two ways: (1) by permitting use of cap screws, instead of bolts and nuts; (2) by allowing use of smaller, shorter, fewer cap screws. *Heli-Coil* inserts *protect* your product from thread wear, galling and stripping for life in every kind of metal, in plastics or wood. They preserve customer good-will by preventing product failure, due to thread fault. *Heli-Coil* inserts improve the end product, cut rejects, salvage threading errors.

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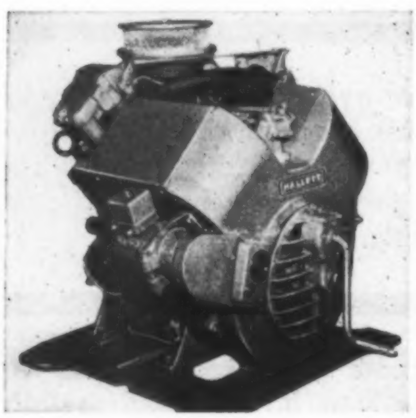
form fiber distribution. The felt is stable to bacteria such as mold or fungi and is highly resistant to acid or alkaline solutions. Made by **American Felt Co.**, Glenville, Conn.

For more data circle MD-66, page 203

Diesel Engine

Two-cylinder Model AC-2 diesel has a four-stroke cycle and is air-cooled with integral flywheel. It is rated 13 hp at continuous load, 14 hp intermittent load. The engine



has a 3½-in. bore and 3¾-in. stroke, 36.07 cu. in. displacement per cylinder and 20 to 1 compression ratio. Produced by **Hallett Mfg. Co.**, Inglewood, Calif.

For more data circle MD-67, page 203

Speed Monitor

In addition to providing accurate indication and control of speed, this monitor will actuate relays and solenoids governing various functions. It operates on mechanical tachometer principle; is unaffected by heat, temperature and moisture

variables; and is driven by flexible shaft and gears from any machine shaft. The unit operates on 115-v 60-cycle single-phase ac and can also be supplied for 35 or 70-v dc. Suggested relay and solenoid applications include gear shifting, operating clutches, visual or audible signalling, temperature, overload and motor-actuated remote valve control. Made by **Jones Motrola Corp.**, Stamford, Conn.

For more data circle MD-68, page 203

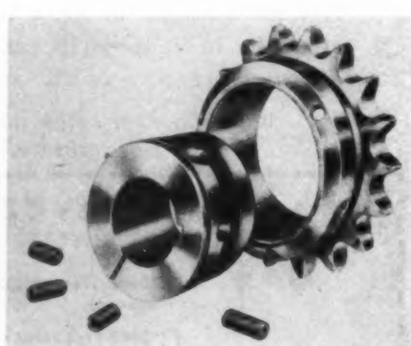
Teflon Sheets

Super-smooth Teflon sheets possessing high tensile strength and measuring approximately 29 by 29 in. are available in any thickness from 0.050 to 2 in. Applications include gaskets, valve seats, diaphragms, and other parts requiring high density and low porosity. Stress-relieved sheets are also available for applications where high dimensional stability is required, particularly at high temperatures. Available from **Ethylene Chemical Corp.**, Summit, N. J.

For more data circle MD-69, page 203

Roller Chain Sprockets

By using different sizes of tapered split bushings or split hubs, Grip-Master roller chain sprockets



eliminate need for boring and simplify changes. Positive press fits are assured. Sprockets are furnished in tapered split bushing and split hub types to meet most requirements. They are completely packaged with mounting instructions. Made by **Cullman Wheel Co.**, 1344 Altgeld St., Chicago, Ill.

For more data circle MD-70, page 203

Limit Switch

Designed for use in explosive gas or vapor-air atmosphere, this explosion-proof, heavy-duty precision switch is listed by Underwriters'

Laboratories as suitable for hazardous locations of Class I, Group C and D. The operating head may be adjusted to any of four horizontal positions, and the roller arm assembly may be reversed to position the actuator roller on either side of the actuator arm. The roller arm operates clockwise and/or counterclockwise. The actuator arm assembly is adjustable through 360 degrees to any of 870 positive lock positions at 0.4-degree intervals. Including the adjustable head, the switch case measures 6 in. high, 2⅛ in. wide and 1 61/64 in. deep. A ⅛-in. thick porous bronze plate fits over the enclosure opening to vent the basic switch cavity, at the same time preventing any great pressure build-up in the cavity due to the ignition of

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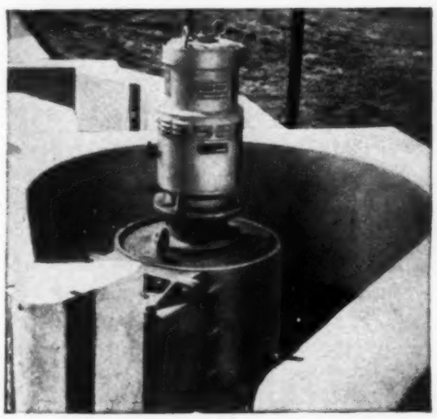

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Chicago Fire Pump, driven by Star-Kimble 40 H.P. squirrel cage motor.

STOP FAST FAST STOP FAST FAST STOP

START SMOOTHLY SMOOTHLY START

FREQUENTLY FREQUENTLY REVERSE FREQUENTLY FREQUENTLY

POSITIVELY POSITIVELY HOLD POSITIVELY POSITIVELY

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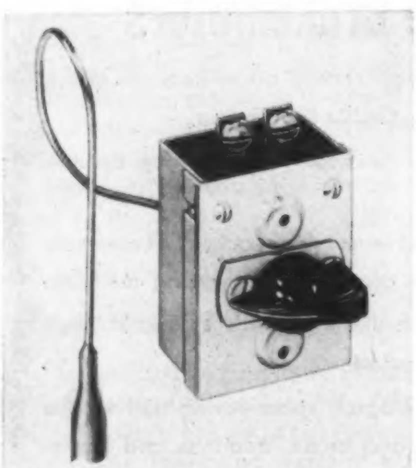
any explosive atmosphere present. The path through the plate also serves to extinguish the flame and cool the hot gases below the ignition point of the surrounding explosive atmosphere. Listed for 20 amp, 110, 220 or 440 v ac; ½ amp, 115 v dc; ¼ amp, 230 v dc; ¾-hp, 110 v ac; 1½ hp, 220 v ac. Pilot duty rating is 125 v ac; 6 amp normal, 60 amp inrush; 250 v ac; 3 amp normal, 30 amp inrush; 480 v ac, 1.5 amp normal, 15 amp inrush; 600 v ac, 1.2 amp normal, 12 amp inrush. Made by Micro Div., **Minneapolis-Honeywell Regulator Co.,** Freeport, Ill.

For more data circle MD-71, page 203

Snap Action Thermostat

In type F Space Saver thermostat, expansion and contraction of stable liquid in bulb actuates contact mechanism by transmitting motion to expansible diaphragm through capillary tube. This motion is multiplied to obtain very close temperature control. Contact mechanism is single-pole, double-

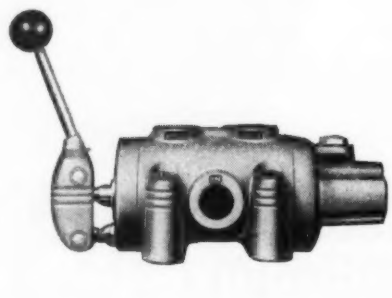


break, snap-action type with silver contacts rated 22 amp at 120 and 240-v ac, noninductive load. Four available units cover 40 to 550 F range. In type F-1, normally-open contacts close on temperature rise, while in type F-2, normally-closed contacts open on temperature rise. Controls can be used with any standard dial or pointer having ¼-in. diameter shaft hole. Made by **Wilcolator Co.,** Elizabeth, N. J.

For more data circle MD-72, page 203

Hydraulic Valve

Four-way valve has ball detents to hold the valve piston in any given position. Locking action of the detent permits vertical as well as horizontal mounting of the valve and prevents movement of the valve piston due to vibration. Valve



is available with one, two or three detents for holding the piston in center position, in either end position, or in both end and center positions. Designed for 1500 psi pressure, the valves are offered for threaded stem, lever, and knob operation; in ¼, ⅜, ½, ¾, 1, 1¼, and 1½-in. sizes and in five piston designs to meet requirements of any hydraulic circuit. Floating piston construction assures close fit of valve piston in valve bore; fit is independent of the fit of the valve stem in the valve covers. Valve stem is supported in its bearings in the valve covers without any influence on the piston fit in the valve body. Made by **Rivett Lathe & Grinder, Inc.,** Brighton 35, Boston, Mass.

For more data circle MD-73, page 203

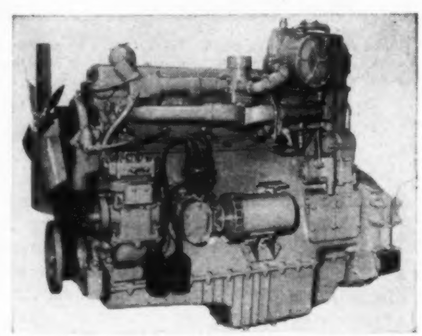
Gasket Stripping

Fluorlastic gaskets consist of long strips of foamed silicone rubber wrapped in sheets of Teflon. Edges of the Teflon extend laterally beyond one side of the silicone core and are heat-sealed to form an impenetrable envelope around the resilient core material. The resulting gasket, which resembles weather stripping, is intended for use in chemical, pharmaceutical, and aircraft industries. Properties are unaffected by temperatures ranging from −130 to 392 F. Made by **Joclin Mfg. Co.,** 2964 Whitney Ave., Hamden, Conn.

For more data circle MD-74, page 203

Industrial Engine

Six-cylinder Model 855 operates on gasoline of various octane ratings or on such LPG fuels as butane and propane. It has a 5½-in. bore, 6-in. stroke, and a displacement of 855 cu in. Operating on regular gasoline, the engine develops a rated gross horsepower of 265 with a net rating of 240 at 2400 rpm, and a maximum torque of 680 lb-ft at 1400 rpm. Using LPG, it develops 295 rated gross horsepower and 276 net at 2400 rpm, with maximum torque at 775 lb-ft at 1200 rpm. The cylinder



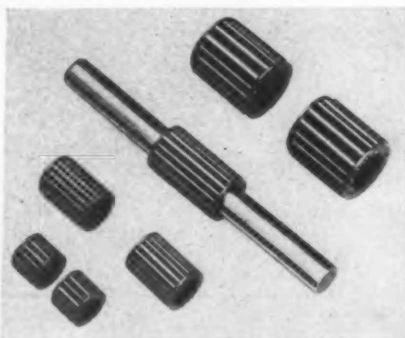
head has a "figure 8" combustion chamber and swirl-type intake ports. Compression ratios are 6 to 1 on the gasoline engines and 9 to 1 on the LPG models. Developed by Hall-Scott Motor Div., **ACF-Brill Motors Co.,** Berkeley, Calif.

For more data circle MD-75, page 203

Roller Bearings

Specially designed for slow moving machinery, line of precision cage type roller bearings is suit-

able for most types of farm machinery, conveyors, farm wagons, wheel toys, material handling equipment. Bearings are statistically controlled for quality and precision in production, hydraulically inspected for uniform stress and

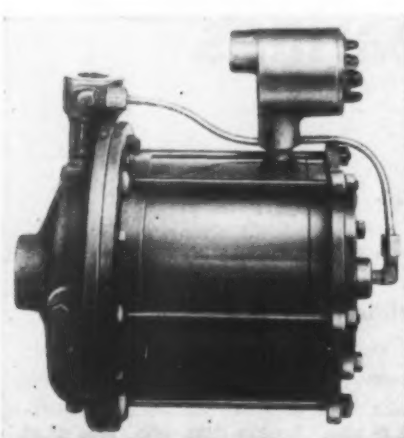


individually inspected for ID and OD specifications. A standard size range and custom models are being produced. Made by **Inland Automatic, Inc.**, Dept. B-104, 1108 Jackson St., Omaha 8, Neb.

For more data circle MD-76, page 203

Centrifugal Pump

The Chempump is totally closed from the atmosphere, sealless and leakproof. It is available in various materials and is used to handle

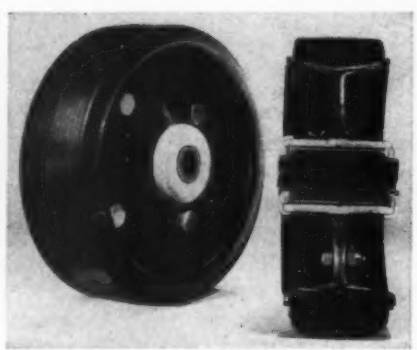


corrosive, radioactive, and highly volatile fluids and for processes employing vacuum. Basic principle of the Chempump permits the fluid being pumped to enter part of the motor. The stator and its windings are isolated from the fluid by a corrosion-resistant, non-magnetic alloy cylinder in the air gap. The rotor is hermetically sealed with the same alloy. Made by **Chempump Corp.**, 1300 East Mermaid Lane, Philadelphia 18, Pa.

For more data circle MD-77, page 203

Industrial Wheel

Model MB has a replaceable tire which is bonded to a steel inner band. Tread does not slip, stretch or creep under heavy loads. The wheel's one-piece hub has a retaining ridge that locks the entire assembly to the wheel disks, preventing shifting under side thrust. Load

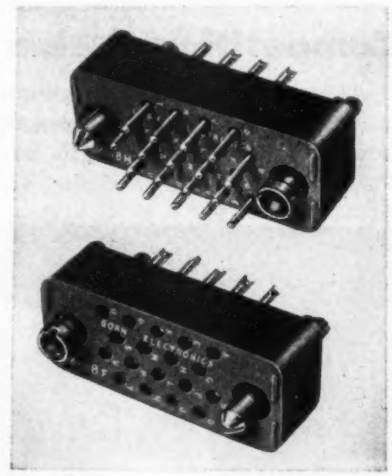


rating is 400 or 500 lb, depending on size. Manufactured in 6 and 8-in. sizes with a $1\frac{7}{8}$ in. width tread in either cushion-rubber or Neoprene by **Rapids-Standard Co., Inc.**, 342 Rapistan Bldg., Dept. MB, Grand Rapids 2, Mich.

For more data circle MD-78, page 203

Miniature Connectors

Compact, lightweight multiple contact precision connectors are for use in both military and civilian industry, such as aircraft, electronic instruments, telemetering, computors, audio and similar applications. Also for portable airborne rack to panel and cable to panel applications. Available with from one to 100 contacts. Contacts are gold plated over silver for maximum

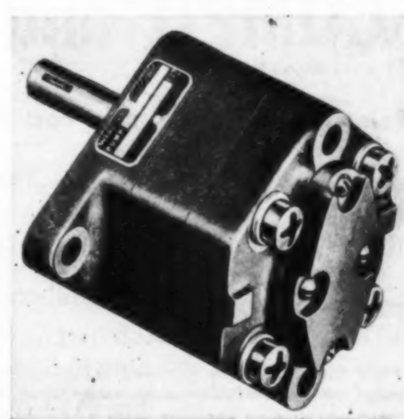


conductivity and corrosion resistance. "Floating" contact design facilitates engaging and disengaging. Connector bodies are made of molded melamine with raised barriers, which provide high resistance to arcing and heat. Connectors are available with 14, 18, 20, 21, 34, 41 and 50 contacts from Gorn Electronics Div., **Gorn Electric Co.**, 895 Main St., Stamford, Conn.

For more data circle MD-79, page 203

Hydraulic Gear Pump

Added to line of hydraulic pumps is this HB series gear type unit in three sizes with capacities ranging from 1.8 to 2.9 gpm at 1800 rpm



and 1000 psi. An internal relief valve is optional and is adjustable for pressure between 500 and 1200 psi. Type HBP model is designed for face or wet sump mounting,

Here's another new modified polystyrene

MC-305 is the easiest to mold of Koppers "high impact" series of Modified Polystyrenes. It has been used in such large area moldings as refrigerator door liners, air conditioning housings, and television masks and tube protectors.

MC-305's high resistance to shock and impact minimizes danger of damage to molded sections during ordinary production line handling as well as during actual use. Its shock resistant qualities also make MC-305 an excellent material for toys and novelties.

Possibly you have a product that can be made better or faster or less expensively with Koppers Modified Polystyrene. To help you choose the right Koppers material for your particular job, we have prepared a new technical bulletin detailing the properties of Koppers Modified Polystyrenes. Write today for your free copy.

KOPPERS POLYSTYRENES

Type 3	General Purpose Polystyrene
Type 7	Improved Heat Distortion Temperature
Type 8	Highest Heat Distortion Temperature
MC-185	High Impact, Lowest Water Absorption Rate
MC-301	High Impact, Improved Heat Distortion Temperature
MC-305	High Impact, Easy Flow
MC-309	High Impact, Highest Heat Distortion Temperature
MC-401	Medium Impact, Improved Heat Distortion Temperature
MC-405	Medium Impact, Easy Flow
MC-409	Medium Impact, Highest Heat Distortion Temperature

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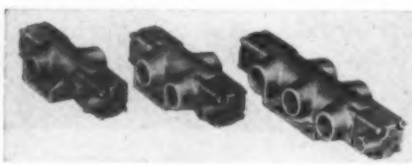
and type HBS is sealed unit recommended for direct, gear or belt drives. Either pipe tap or manifold porting is available. Applications include industrial trucks and power steering systems, tractor loaders, machine tools and construction machinery. Made by Oil Hydraulics Div., **Webster Electric Co.,** 1900 Clark St., Racine, Wis.

For more data circle MD-80, page 203

Air-Hydraulic Controls

Utilizing same hand, cam, pilot or foot actuating devices as employed in previously announced 1/8 and 1/4-in. NPT sizes, larger 3/8 and 1/2-in. NPT pneumatic and hydraulic valves are offered in two, three and four-way types. Readily dis-

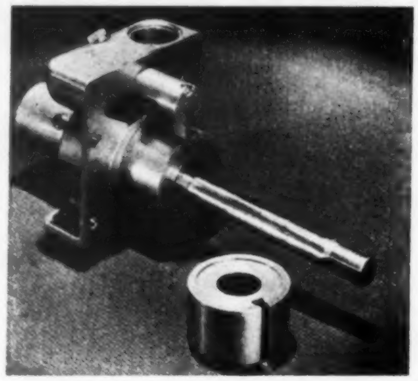


assembled for servicing with screwdriver without disturbing connections, controls feature nonextruding O-ring seals, balanced-pressure spools, and full-flow design. Pressure range is up to 200 psi. Made by **Versa Products Co. Inc.,** 249 Scholes St., Brooklyn, N. Y.

For more data circle MD-81, page 203

Impeller Pump

A bronze-impeller, positive-displacement pump handles light or heavy oils and greases. Maximum capacity is 10 gpm. Two axially oscillating impellers deliver continuous, nonsurging flow without foaming or air entrainment. The precision machined and sintered bronze powder-metal impellers of the pump act like oilless bearings operating in a lubricating medium, in that they absorb and retain oils, minimizing the dangers of heat seizures and scorched impellers. The pump is self-priming in low temperatures, is suitable for oper-

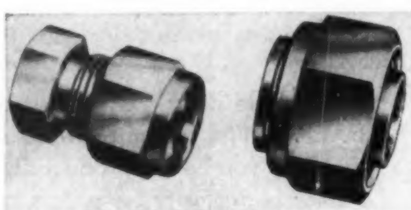


ating oil temperatures up to 500 F. It operates submerged and will pump in either direction and in any position. **Eco Engineering Co.,** Newark, N. J.

For more data circle MD-82, page 203

Cap and Plug Fittings

The cap, left below, is used for capping end of tube; the plug, right below, is used for plugging unused ports of Swagelock fittings. Both come completely assembled and require only 1 1/4 turns to provide a torque-free, leakproof seal. Caps and plugs are available in various sizes from 1/8-in. up in brass, aluminum, steel, stainless

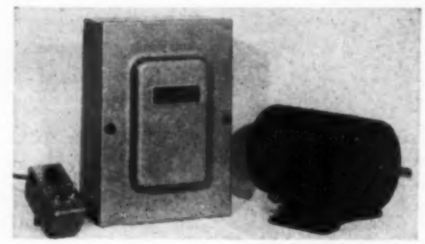


steel and Monel, from **Crawford Fitting Co.,** 884 East 140th St., Cleveland 10, O.

For more data circle MD-83, page 263

Variable Speed Control

Electronic servospeed system controls speed of series or universal motors and fractional and subfractional horsepower motors. A speed range of as much as 100 to 1 is achieved with motors having a base speed of 5000 rpm or higher. Three wire motors may be controlled for one direction of rotation, and four wire motors may be operated in either direction by use of a small reverse switch. An electronic servo circuit provides smooth, stepless speed control and employs a composite torque-speed feedback signal. Motors are available in straight motor and gearmotor types from 1/100 to 1/10-hp for operation on 115-v, 50-60 cycle lines; 220 v models are available up to 3/4-hp. Models are made



without cabinets for incorporation into machines or with cabinets for bench or wall mounting, by Servospeed Div., **Electro-Devices Inc.,** 4-6 Godwin Ave., Paterson, N. J.

For more data circle MD-84, page 203

Seals

Available in shaft sizes ranging from 1/4 to 2 3/4 in., thin rind seals for Torrington needle bearings fit the same bore as the bearing itself, eliminating a counterbore in the bearing housing. The seals, made from Goodyear Hycar rubber, insure proper lubrication of the bearings and resist the deteriorating effects of oils, greases and high-

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You're looking behind the scenes at the American Locomotive Company, Ordnance Division, Schenectady, New York.

The prints you see streaming from the processing machine are positive photographic reproductions of M47 tank drawings. They're produced without a negative step . . . and under ordinary room light.

This fast, low-cost operation enables the Ordnance Division to protect thousands of M47 drawings against expensive wear and tear. For the drawings are filed away after the Autopositives are made. Then, because the engineering department of the Ordnance Division holds the design contract for the M47, these highly legible and durable intermediates are used to produce all the quantity prints required by the 29 manufacturers and government agencies involved in the distribution of engineering changes on the M47 tank.

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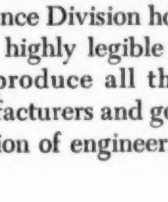

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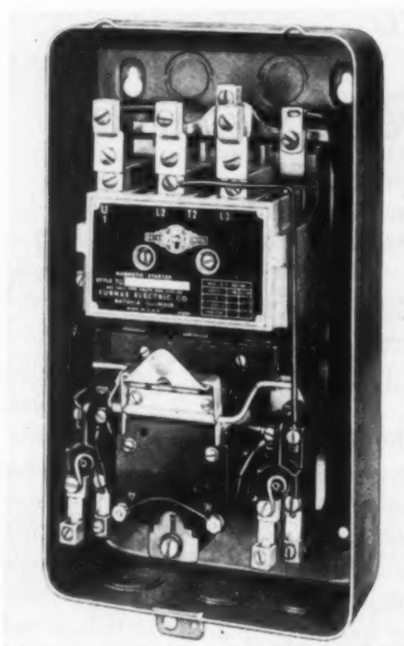
speed abrasion. They also have good permanent set or distortion resistance. Seals are precision-molded to the same sizes as the bearings, with close tolerances of ± 0.002-in. OD to $\pm$ 0.005-in. ID. Made by **Albert Trostel Packings Ltd.**, Lake Geneva, Wis.

For more data circle MD-85, page 203

Magnetic Starters

Neither conventional size 2 nor size 3, these intermediate controls have a capacity between the two sizes and are designated size 2½ by the manufacturer. The controls, which include reversing types, are

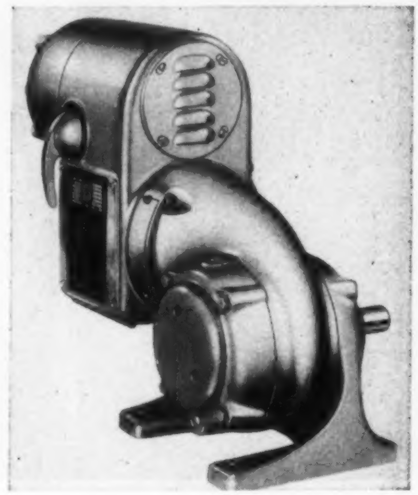


rated 20 hp at 208-220 v, 30 hp at 440-550 v ac polyphase. Some are available with built-in pushbuttons or selector switch. Made by **Furnas Electric Co.**, 1045 McKee St., Batavia, Ill.

For more data circle MD-86, page 203

Variable-Speed Drive

Offered in ¼, ⅓, ½ and ¾-hp sizes with speed variations up to 10 to 1, type VA-GW Varidrive-Syncrogear has low variable speeds

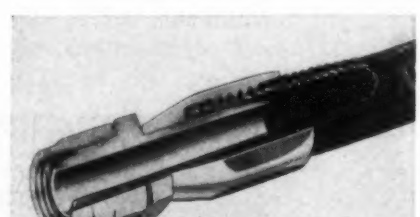


with a right-angle hook-up for such applications as conveyor belts. Full speed range is selected by turning control handle through an arc of 300 degrees. Because the variable-speed transmission is of the V-belt type, speed changes as small as 1 rpm can be effected while the motor is operating. Construction features include automatic drive belt tensioner, asbestos protected windings and normalized castings. Made by **U. S. Electrical Motors Inc.**, P. O. Box 2058, Terminal Annex, Los Angeles 54, Calif.

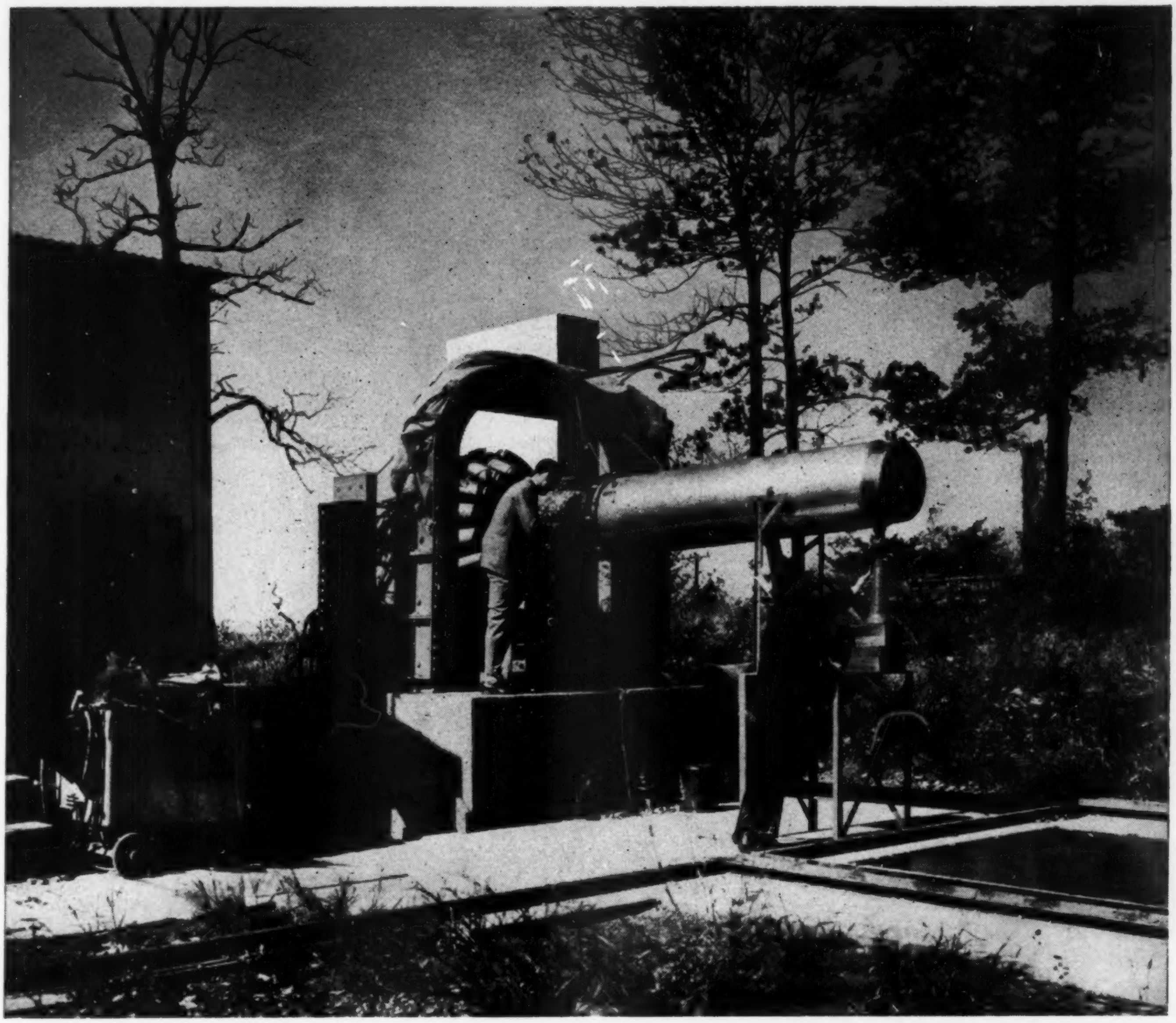
For more data circle MD-87, page 203

Hose Lines

Globe-Seal lines can be connected to Ermeto and Ferulok type flareless tube systems, to which they mate without ferrules or inserts.



The lines, however, have detachable, reusable fittings. Assembling the fitting to flexible hose can be accomplished with ordinary hand tools. Hose line is available in both single and double wire braid

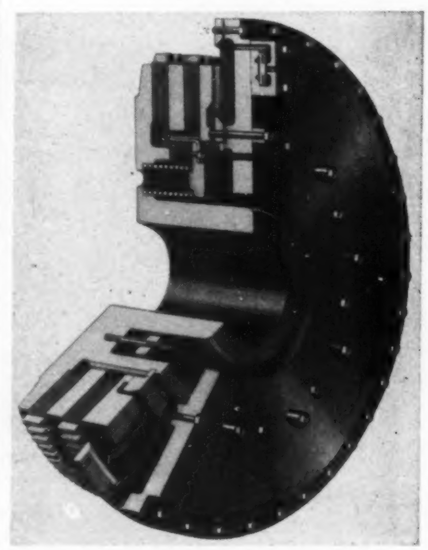


types from **Aeroquip Corp.**, Jackson, Mich.

For more data circle MD-88, page 203

Air-Actuated Clutches

Developed from previous models, the PO air-actuated clutches have a higher torque capacity, and are narrower and lighter. The new design permits units to be installed in

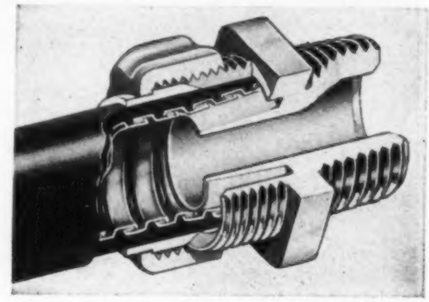


less shaft space, with closer shaft-bearing center distances. Air actuation is effected through insulated and air-cooled diaphragms. All parts are fabricated from specially developed alloys, for maximum strength to withstand heavy load shocks and tooth wear. Made by **Twin Disc Clutch Co.**, Racine, Wis.

For more data circle MD-89, page 203

Conduit Fitting

This liquid-tight fitting mates with any conduit spiral and is installed without being disassembled. Made in straight, 90 and 45-degree elbow designs, it accommodates conduit from ⅜ to 2 in. trade size. A color-coded plastic seal in the gland indicates size of fitting and the fact that it is a liquid-tight type. Raceway is properly grounded when colored seal is visible after

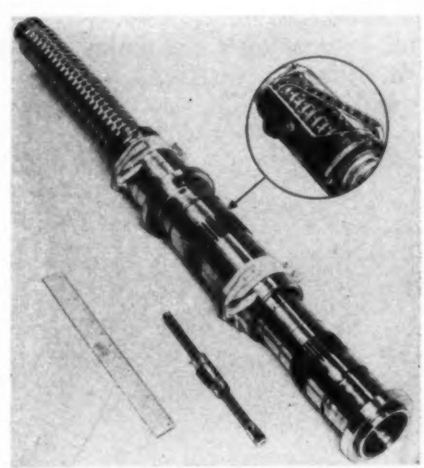


installation of the fitting. Fitting body in three smallest sizes, is die cast from a high-strength zinc alloy; other four sizes are of malleable iron. The conical ground connection is wedged inside the conduit, and the gland is tightened around the conduit to complete the joint. Made by **Thomas & Betts Co.**, Elizabeth, N. J.

For more data circle MD-90, page 203

Ball Screwjack

Utilizing physical principles of screw expansion and the near-frictionless spacer ball, coupled with tolerances up to 0.0004-in. in the ball bearing races, this screwjack is made with a mating nut and screw having helical ball races precision ground to highly accurate tolerances. The ball circuit is



closed with a return tube which is attached to the nut, allowing the ball bearings to circulate freely and resulting in the transmission of rotary motion to linear motion with

a small friction loss. Available in sizes from 3/8 to 4 in. diameter, up to 22 ft long from **Vard Inc.**, Pasadena, Calif.

For more data circle MD-91, page 203

Synchronous Timing Motor

Oil retaining disks which trap oil at bearing surfaces for leakproof operation are feature of series 50 synchronous timing motor with torque rating of 40 oz-in. at 1 rpm. Other advantages include cut gears and pinions throughout, polished spindles; 50 per cent more bearing surface than previous models, and positive positioning and locking of stator and rotor assemblies. Motors are available in unidirectional or reversible models in wide variety

of output speeds from 1/6 to 1800 rpm and for operating voltages from 6 to 550 v ac. Made by **Hagen Mfg. Co.**, 202 Twentieth St., Moline, Ill.

For more data circle MD-92, page 203

Needle Valve

This all-steel valve is designed for working pressures up to 10,000 psi. Body and valve stem guide are machined from heavy bar stock steel and are joined by the Conoweld process. This integral unit provides a positive leakproof joint

and eliminates the danger of the valve stem guide becoming loose while unscrewing the valve stem. Deep chambered threads are machined to extra long length, permitting even undersized pipe threads to make up tightly. Valve stem of 416 stainless steel with fine pitch threads permits close throttling while retaining the strength necessary to withstand twice the normal working pressure. Marpak packing is used; valve body, packing gland and packing nut are electro zinc plated to prevent corrosion; cross type handle is of malleable iron finished in heavy baked enamel. Developed by **Jas. P. Marsh Corp.**, Skokie, Ill.

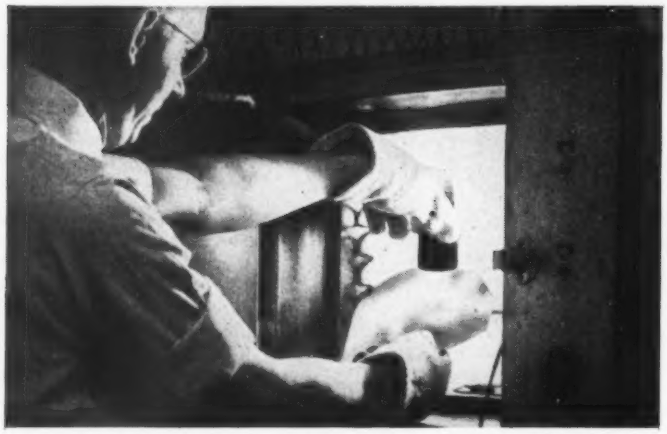
For more data circle MD-93, page 203

Beryllium Copper Strip

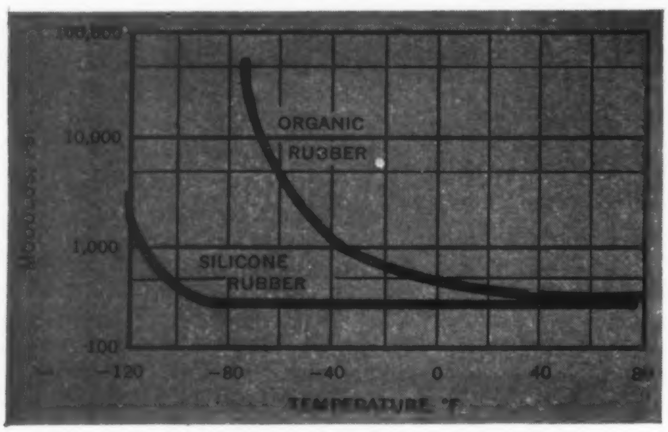
Alloys such as No. 10, 25 and 165 are now supplied in strip up to 8 in. wide and as thin as 0.0005-in. to ±0.0001-in. tolerances and either solution annealed or rolled to temper. Typical uses include current-carrying springs, switches and circuit breakers; ultrasensitive pressure diaphragms and bellows and other pressure - responsive units; corrosion-resistant bushings and bearings; and pump, valve and

(*Continued on Page* 232)

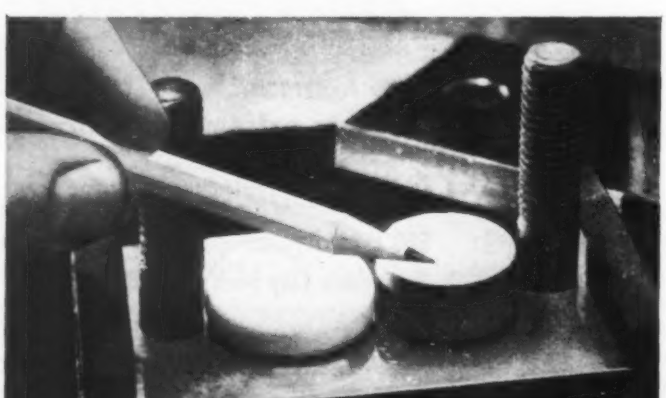
REMARKABLE DESIGN NEW SILICONE DEMONSTRATED BY TEST...



High temperature resistance: Elevated temperatures have no appreciable effect on the 1000-1200 volts per mil dielectric strength of glass cloth coated with SE-100.



Flexible at −120 F: A glance at the blue curve shows the difference (Young's Modulus in flexure) of SE-550 versus organic rubber after 24 hours.



Compression test: Same-size pieces of ordinary silicone rubber and SE-360 are compressed for 22 hrs. at 350 F. Note how SE-360 (right) "comes back" to original size.

SE-100 —IDEAL FOR ELECTRICAL AND MECHANICAL APPLICATIONS!

General Electric's new silicone rubber coating compound, SE-100, combines outstanding heat resistance, electrical and physical properties for a wide variety of electrical and mechanical applications. SE-100 may be coated on glass or organic fabrics for service at high or low temperatures or where resistance to weather, ozone, corona or chemicals is required.

SE-550 —STILL FLEXIBLE AFTER 24 HOURS AT —120 F!

G. E.'s new extreme low-temperature silicone rubber, SE-550, combines high strength and elongation with maximum low-temperature usefulness. SE-550 shows practically no increase in modulus at −100 F and retains useful flexibility at −120 F. This flexibility is achieved without sacrifice of high-temperature resistance or any of the other desirable properties inherent in silicone rubber.

SE-360 —MORE "COME BACK" THAN ANY KNOWN RUBBER!

G. E.'s new low compression set silicone rubber, SE-360, is designed to provide more positive sealing action in parts subject to compression at elevated temperatures. In addition to its outstanding low compression set, SE-360 has unusually low shrinkage when cured. This means parts with more uniform properties, closer tolerances and opportunities for your fabricator to cut scrap loss.

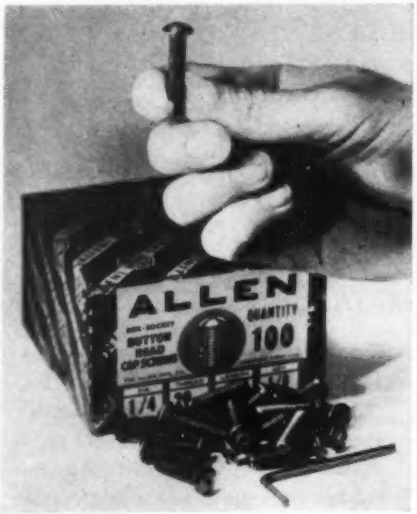
(*Continued from Page* 229)

other engine parts. Strip is processed by Industrial Div., **American Silver Co. Inc.,** 36-07 Prince St., Flushing 54, N. Y.

For more data circle MD-94, page 203

Button-Head Screws

The rounded tops and flush edges of these fasteners provide a smooth surface where countersinking is not possible. Manufacture of the screws employs the Pressur-Forming process, in which the screw body is cold-worked and the socket head is cold-drawn to impart extra strength to the metal by keeping the steel fibers continuous and uncut. Sizes range from No. 8 by ¼-in. through ⅝ by 2-in. standard with NC threads. Same sizes are also standard with NF threads, except in ½ and ⅝-in. diameters. Made by **Allen Mfg. Co.,** Hartford, Conn.



For more data circle MD-95, page 203

Breather Plug

Although specifically designed for use in refrigerator and freezer units, plug may be used to cover any hole in sheet metal or plastic units and still provide for air passage or easy access. Installation is

accomplished by pressing the plug into the hole with the fingers. As the three spring legs enter the hole, they depress; once through the hole, the legs spring back and hold the plug securely. Available in sizes to fit ⅞ and 1-in. holes from **Shakeproof Div.,** Illinois Tool Works, St. Charles Rd., Elgin, Ill.

For more data circle MD-96, page 203

Fiberglas Cloth

Woven of Fiberglas yarn, this cloth measures 0.001-in. thick and weighs 0.81-oz per square yard. It is used for electrical apparatus as the carrying medium for mica and insulating varnishes, permitting reduction in size of electrical apparatus or for general plastic and film reinforcement. Cloth is produced by **Soule Mill,** New Bedford, Mass.

For more data circle MD-97, page 203

Panel Meter

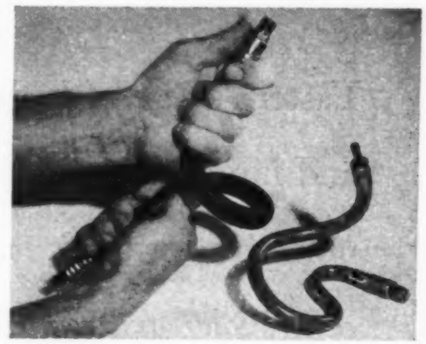
Designed to avoid reading errors, the Multi-Dialer exposes only one scale to view at a time and, at the same time, automatically switches the appropriate internal or external circuitry associated with that

scale. It is available with five self-contained ranges in volts, milliamperes, microamperes and amperes, or custom-engineered to specific equipment. Used as a panel instrument, it is mounted to standard JAN 3½ in. dimensions. Made by **Marion Electrical Instrument Co.,** 400 Canal St., Manchester, N. H.

For more data circle MD-98, page 203

Coolant Tube

Extremely flexible, this copper-core tubing can be twisted into almost any shape. It remains in the twisted shape until reshaped. Thus,

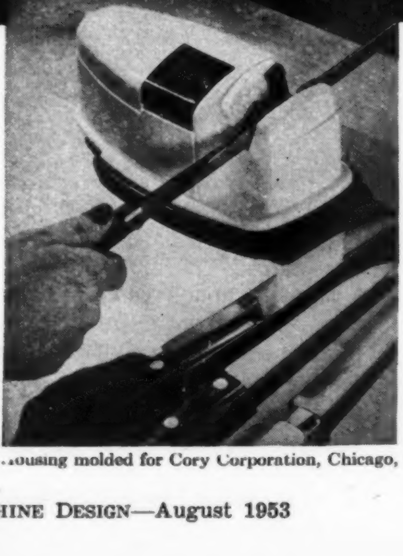


coolant flow is uninterrupted, and since tubes do not need to be wired or taped into position, the coolant can be directed into otherwise inaccessible places. Tubes are made of Neoprene rubber and are available in most diameters and lengths and with any length tip from **Acrobat Co.,** 1923 Vineyard, Los Angeles 16, Calif.

For more data circle MD-99, page 203

Pressure Switch

Heavy-duty Model 9612 is capable of sensing any system pressure over an adjustable range from 15 to 3000 psi and actuating an electrical circuit on increasing or decreasing pressure. Piston type pressure sensing element is not subject to fatigue. Basic switch may be converted to changing service requirements by means of inter-



Housed in Hercocel A, the new Cory Electric Knife Sharpener has everything that makes for economical production and profitable sales. The gleaming white, attractively styled housing is quickly molded in one piece. It's tough, dimensionally stable . . . proof against casual kitchen handling. The porcelain-like finish, with color through and through, is stain-resistant . . . can't chip, peel or wear off. And Hercocel, non-resonant itself, helps assure quiet operation of the unit. For complete information on Hercocel plastics, and details of the design and technical assistance offered by Hercules, write:

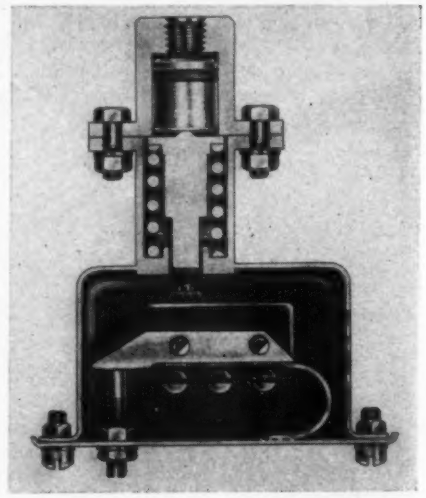
HERCULES POWDER COMPANY
INCORPORATED
Cellulose Products Department, 950 Market St., Wilmington 99, Del.

Housing molded for Cory Corporation, Chicago, Ill., by G-M Laboratories, Inc., Chicago, Ill.

CP53-7

New Parts

changeable fitting kits. The mounting bracket which fits around the neck of the unit may be rotated a full 360 degrees as well as moved up and down. Switch may be

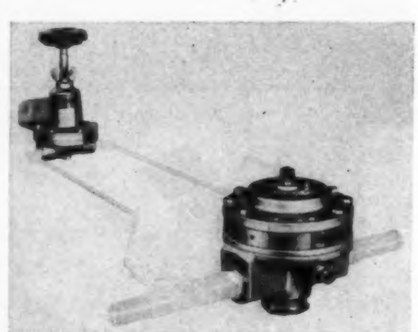


mounted in any position and is not sensitive to vibration or line surges. Made by **Barksdale Valves,** 5125 Alcoa Ave., Los Angeles 58, Calif.

For more data circle MD-100, page 203

Air-Pressure Regulator

Pilot-controlled, this device utilizes regulated air pressure to control delivered air pressure, minimizing pressure fluctuations. Operating range is from 2 to 120 psi, with



a capacity to deliver more than 200 cfm. Design of regulator allows placing it where it will be most effective and installing the separate pilot control regulator where convenient. Available in ½, ¾, and

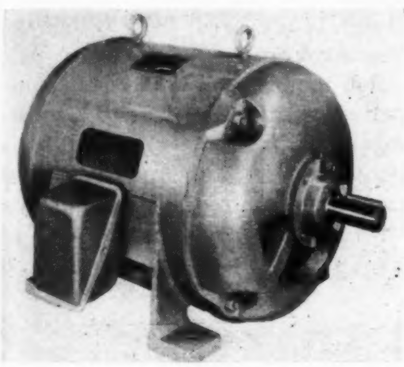
New Parts

1-in. sizes from the **C. A. Norgren Co.,** 3442 South Elati Street, Englewood, Colo.

For more data circle MD-101, page 203

Motor Frames

Electric motor frames of the T-type design, in ratings up to 250 hp, have been modernized. These motors, from frame size 444 to 586 of drip-proof construction, have rolled-steel frames and cast-iron end bells, assuring permanent alignment. Bearing caps at the shaft ends give



longer bearing life. Made by Electro Dynamic Div., **General Dynamics Corp.,** Bayonne, N. J.

For more data circle MD-102, page 203

Silicone Rubber Insulation

Possessing a combination of physical toughness and electrical strength undiminished over a wide temperature range, SE-100 silicone rubber coating can also be used for heat resistant engine gaskets, flexible heater ducts and other nonelectrical products. The putty-like material is expected to find application for class H insulation in electrical equipment used under hot or moist conditions. Dielectric strength is retained after exposure to temperatures as high as 315 C. It can be coated on glass tapes, glass wrappers and other insulating materials at low cure temperatures. Made by **General Electric Co.,** Chemical Div., Plastics Ave., Pittsfield, Mass.

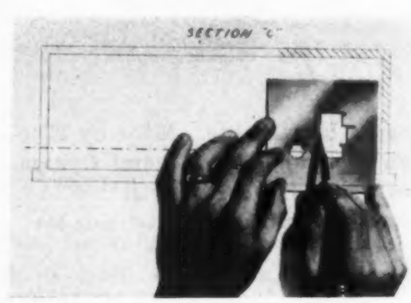
For more data circle MD-103, page 203

ENGINEERING DEPARTMENT

EQUIPMENT

Drafting Template Cutter

Draftsmen and engineers may eliminate the process of redrawing standard elements each time they appear in a drawing by employing plastic templates cut to suit individual drafting needs. A master design is prepared, from which the Engravograph reproduces a precision template on a sheet of transparent plastic by means of a rotating engraving cutter guided by a pantograph. The engraving cutter can be shaped to provide a template with beveled edges conforming to



the angle of the pencil point necessary to produce accurate lines. For firms not wishing to cut their own master templates, the manufacturer will furnish the templates from drawings or sketches submitted. Manufactured by **New Hermes Engraving Machine Corp.**, 13-19 University Place, New York 3.

For more data circle MD-104, page 203

Portable Photocopier

Compact size—small enough to fit into a desk drawer or a briefcase—yet ability to copy anything up to 8½ by 14 in. in size are the outstanding features of the "Contoura"—a portable photocopy ma-

ELIMINATE

waste motions in production

SWITCH TO

Eaton Springtites and Sems

(Bolt or screw preassembled with spring lock washer)

The motions used by a production line worker on bolted assembly operations shown in figure 1 above differ from the motions used by the worker pictured in figure 2. This difference is graphically shown by light lines taken from actual factory production photographs. This difference in motions made possible by Eaton Springtites and Sems means higher production quotas and increased manufacturing profits. Why?

Because they are preassembled units, Eaton Springtites and Sems eliminate the need for assembling the spring lock washer on the bolt. Losses of separate washers and bolts are cut, while double requisitioning, purchasing, inventory is eliminated. Then too, Eaton Springtites and Sems are easy to use in hard-to-get-at places, and yet can be hopper fed for automatic screw driving. Most important, Eaton Reliance quality preassembled Standard ASA Spring lock washers and bolts assure greater product quality. See your nearest Reliance Sales Office for complete details on Eaton Springtites and Sems or write directly for Engineering Bulletin No. S-49.

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chine. Unit will allow copying of curved pages in magazines and books without harming the binding and without removal of the page being copied.

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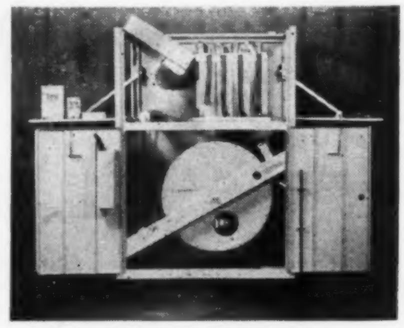
For more data circle MD-105, page 203

Oscillograph Film Processor

New portable photographic processing machines that automatically develop, fix, and dry either oscillograph film or paper may be loaded and operated in daylight. Self-contained, standard models deliver 3 to 5 ft of completely developed and dried paper per minute. A larger model delivers up to 10 ft per minute. Standard models are also available for processing film in 16 mm, 35 mm, 70 mm, and 12-in. sizes at speeds up to 5 ft per minute. Readily portable, the machine rolls on swivel-mounted rubber-tired casters. Processor operates



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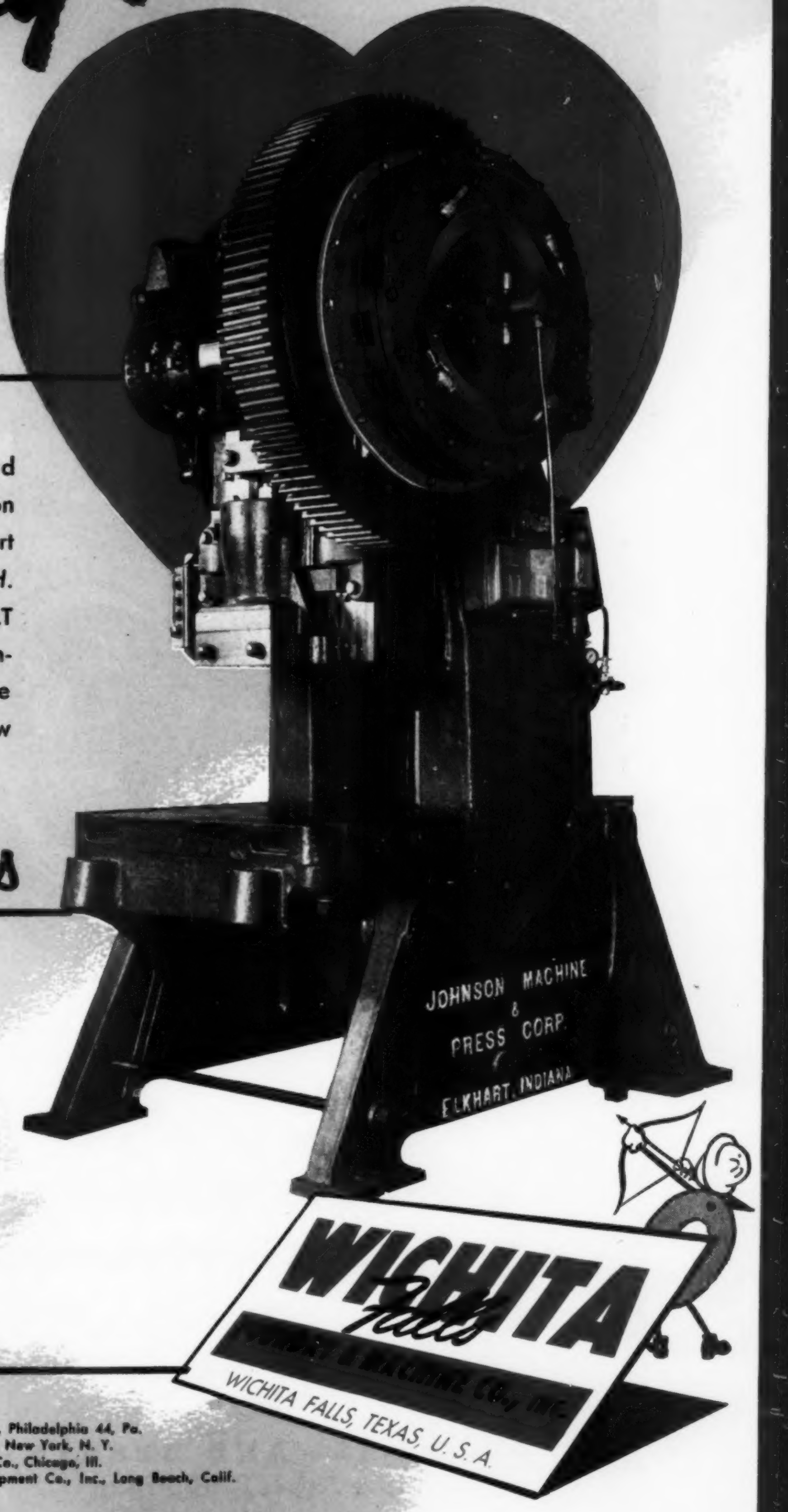

For smoother, faster, more consistent and trouble-free operation, this Johnson 90-ton OBI press was equipped with a Strong Heart ...*The WICHITA Low Inertia Air-Tube CLUTCH.* This clutch is compact, powerful, and BUILT FOR YEARS OF SERVICE. Make your machinery more efficient with increased performance and production by changing to Wichita Low Inertia Air-Tube Clutches...NOW!

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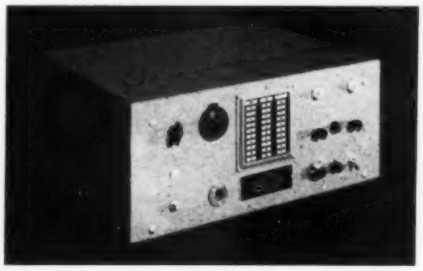
For more data circle MD-106, page 203

Analog Converter

Analog voltages are automatically converted into decimal digits with an accuracy of 0.1 per cent (1000 counts full scale) by type 24A Teleducer. Instrument operates upon demand, digitizes an input voltage and holds the representation for controlled periods of time for purposes of display, recording or any readout form. Output can be recorded by means of



punched cards, electric typewriter, magnetic tape, or punched tape. Instrument digitizes low voltage without direct current amplification and high voltage by means of attenuators. Developed by **Telecomputing Corp.,** 133 E. Santa Anita Ave., Burbank, Calif.

For more data circle MD-107, page 203

Strain Gage

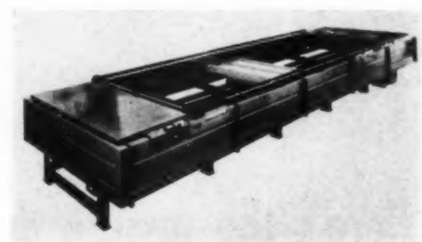
The smallest SR-4 bonded resistance-wire strain gage yet developed by Baldwin - Lima - Hamilton has been added to their line of more than 100 sizes and types of these gages. This new type AB-32 strain gage has a 1/32-in. gage length, which is half that of the former smallest gage. Made with a Bakelite base of the wrap-around type with cupro-nickel wire, it is applied with phenol-resin cement. Nominal resistance is 120 ohms and average gage factor is 1.4, which is guaranteed within 10 per

cent. Two pairs of leads are available and grid wire is only 0.0005-in. in diameter. Made by **Baldwin-Lima-Hamilton Corp.,** Philadelphia 42, Pa.

For more data circle MD-108, page 203

Large Size Printer

Completely self-contained except for lamp transformers, this large size printing frame is available in various sizes up to approximately 8 ft wide and 20 ft, 10 in. long. Unit exposes all light-sensitive materials and has a variable-speed light source that provides uniform light distribution by means of a complete focal plane sweep. All operations are performed without any manipulation of frame.



Main light source consists of one or more quartz mercury-vapor arc lamps. Drive member is an electronically controlled, variable-speed dc motor with built-in worm reducer gear. The contact is established between a vacuum blanket and plate glass. Manufactured by **Charles Bruning Co. Inc.,** 4700 Montrose Ave., Chicago 41, Ill.

For more data circle MD-109, page 203

Counter and Timer

Model 5500 provides a direct reading of elapsed time between any two events, or of the number of events occurring during a specific interval; a means of measuring low frequencies and a straightforward electronic counting facility. Elapsed time from 40 microseconds to 100,000 sec can be meas-

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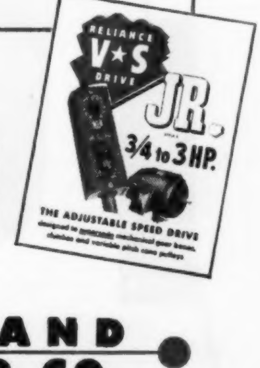
Ask for Bulletin D-2102. It describes and illustrates features, applications, components and operation; dimensions and characteristics also are included.



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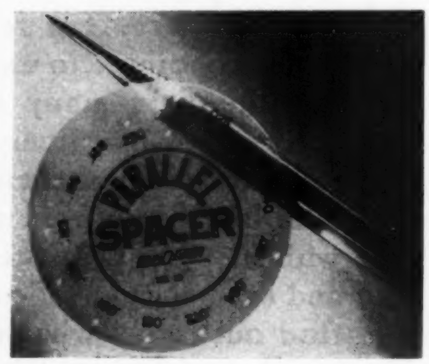
ured directly with an accuracy of ± 10 microseconds, and events occurring at rates from 20 to 100,000 per second can be counted. Features scanning to any mutiple of 1 or 10 sec, direct test of all time base frequencies, automatic operation with display from 0.5 to 5 sec, and manual operation with infinite display. Made by Berkeley Scientific Div. of **Beckman Instruments Inc.**, 2200 Wright Ave., Richmond, Calif.

For more data circle MD-110, page 203

Parallel Spacer



Unit is designed to draw lines from 0.025 to 0.375-in. distance from practically any smooth reference curve. Provision is made for 14 gradations of distance from the reference curve. The point of a hard pencil is inserted in the selected hole, and the tracer is guided along the edge of a straight or curved figure. Uses include graph plotting, cross-hatching, lettering, etc. Made of 0.030-in. thick mathematical quality plastic, the template has an actual diameter of 2 in. Available from **Rapidesign Inc.**, P. O. Box 592, Glendale, Calif.

For more data circle MD-111, page 203

Stress Relief

OPPORTUNITY makes appearances in many guises. J. P. Henderson exposes one here.

You Can't Tell Them Everything!

Many times in hiring a new man for a job, or transferring an old employee to a new position, the boss is unable to explain all of the opportunities involved. There is a variety of reasons why this is so.

Let's consider an example.

Some years ago, I hired a young man for a specific job. He was to work with several other men under a supervisor who appeared hale and hearty, with no chance of dying off soon enough to provide an opening ahead. I explained to him that there was a good opportunity on this job and that if he did well, there were several places in the organization to which he might look for advancement. As for him, he probably thought it was the usual old stuff, the inspirational speech that every boss hands out, hoping to get as much work as possible out of the employee.

Actually, the story was this: After culling over all of its likely prospects, another division of the company decided they would need a pushing young fellow to head up a new department. They wanted him trained in my department for about two years as background, after which he would transfer and get further training in that plant.

I had no extra man at the time and several likely prospects had been approached but did not want to move. So there it was, advancement all planned; a pushover for an ambitious young fellow who would have an excellent job in from four to five years.

But I could not tell him all of that. I could not commit myself on that whole program to this green young man for several reasons.

At the end of a year or two, we would have considerable money and time invested in his training. Suppose it became obvious by then

Stress Relief

that he was not the executive material we hoped for. We could still use him elsewhere in the organization and not lose the investment in his training. But if he had been promised the other job, he would naturally have been dissatisfied with this later arrangement.

Furthermore, suppose I had revealed to him this entire program. It's possible he might have exerted himself abnormally to secure the advancement and then eased off. Actually I wanted to see his normal drive without a definite bait in front of him.

Cases differ in details, but the same sort of thing can often happen.

When you are sitting across from the boss's desk being told about a new job and hearing him talk about opportunities if you work hard, etc., etc., are you cynical enough to think that it's the same old guff, and that he is just trying to lead you on to work your head off? More often than not, the thoughts in the back of the boss's head are something like this: "I sure hope this is the man we're looking for. I hope he has the drive we want and that he can make good. There are so many spots where we can use a smart young fellow—let's hope he's it."

—J. P. Henderson

They Say . . .

"Modern industrial progress depends on teamwork to such a degree that the graduate must be able to submerge his individuality to a considerable extent in order to become an effective member of that team. As a member of the team, he must be capable of making his own contributions, yet be flexible enough to change his own ideas as new thoughts are presented to him by other members of the group. Few indeed are the industrial positions wherein an engineer can work as a 'lone wolf.' He must learn to work with people until his responsibilities increase to such an extent that the major requirement is for people to work with him."—John F. Gordon, *vice president, General Motors Corp.*

Meetings

AND EXPOSITIONS

Aug. 19-21—
Western Electronic Show and Convention to be held at Civic Auditorium, San Francisco, Calif. Additional information may be obtained from society headquarters, Merchandise Mart, San Francisco 3, Calif.

Sept. 6-11—
American Chemical Society. Fall meeting to be held at Hotel Conrad Hilton, Chicago, Ill. R. M. Warren, 1155 16th St., Washington, D. C., is assistant secretary.

Sept. 13-17—
Electrochemical Society Inc. Fall meeting to be held at the Ocean Terrace Hotel, Wrightsville Beach, N. C. Dr. Henry B. Linford, 235 West 102nd St., New York, N. Y., is secretary.

Sept. 20-23—
Packaging Machinery Manufacturers Institute. Twenty-first annual meeting to be held at the Skytop Lodge, Skytop, Pa. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

Sept. 21-22—
Steel Founders' Society of America. Fall meeting to be held at the Homestead, Hot Springs, Va. Additional information may be obtained from society headquarters, 920 Midland Bldg., Cleveland, O.

Sept. 21-25—
Instrument Society of America. Annual meeting and joint conference with the Industrial Instruments and Regulators division of ASME to be held concurrently with the eighth national instrument exhibit at the Sherman Hotel, Chicago, Ill. Richard Rimbach, 921 Ridge Ave., Pittsburgh, Pa., is exhibit manager.

Sept. 28-30—
Association of Iron & Steel En-

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BECAUSE . . . it's the only clutch with SEPARATOR SPRINGS that prevent heating in both HORIZONTAL and VERTICAL installations.

For some time there has been a demand for a single disc clutch for positive control of power transmission in the field of light machinery . . . a clutch that would stay cool on the job in both horizontal and vertical installations. This is "it."

If extreme compactness is not an essential, the Johnson No. 350 (3 h.p. max.) and No. 450 (6 h.p. max.) will meet or beat competition . . . as a better clutch at a competitive price. They incorporate most of the design principles of our Maxitorq multiple disc . . . and except for compactness and capacity, you will find them tops in their field. (Please read column at right . . . "Frankly Speaking" . . . for use recommendations.)

Several driving combinations are available, including V-belt. A simple hex key frees the knurled ring for easy manual adjustment. The "floating" disc prevents drag, abrasion, and heating in neutral. Machine and product designers will solve many problems in the light machinery field with this new Johnson single disc Clutch.

Send for Bulletin No. 250-MD-8

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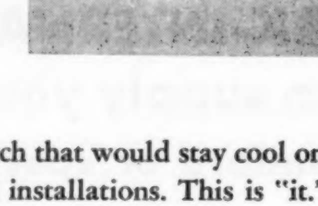

Frankly SPEAKING

Because of a large backlog of orders, The Carlyle Johnson Machine Company found it necessary to postpone the announcement of the new Johnson Single Disc Clutches.

Now, with added equipment for high speed production, it is possible to produce and ship the new clutches without undue delay. In fact, small orders or units for tryout in new machines will be forwarded at once. Design features make the Johnson especially suitable for installation in the following machinery:

Accessory Drives, Air Compressors, Bag Making, Boat Drives, Bread Wrapping, Chain Drives, Combines (farm), Conveyors, Crop Seeders, Cultivators, Dusting Machines, Feed Grinders, Floor Scrubbing, Fruit Cleaners, Gasoline Engines, Generator Drives, Hay Balers, Hoists, Loaders, Lawn Mowers, Milk Coolers, Mixing Machines, Motor Scooters, Packaging Machines, Paper Shredders, Power Fans, Power Saws, Power Take-offs, Pumping Equipment, Sand Spreaders, Sewing Machines, Sheep Shearers, Spraying Equipment, Textile Machinery, Threshers (small), Tobacco Machinery, Tool Grinders, Tractors (garden), V-Belt Drives, Vegetable Sorters.

Naturally, these are but a few of the possible applications for the Johnson Single Disc Clutch. The field is wide open, with new machinery constantly being developed.

Included in the driving combinations are: Gear Tooth, Bolted Plate, Pulley Type, Hub Adapter, Cut-off Coupling Adapter, Single V-Belt Pulley Drive, and Double V-Belt Pulley Drive.

Carlyle Johnson engineers offer their engineering assistance in cooperation with your engineers and machine designers to develop the correct solution of your power transmission requirements. Write to Frank R. Simon, The Carlyle Johnson Machine Co., Manchester, Conn.

Advertisement

Meetings and Expositions

gineers. Annual meeting to be held at Hotel William Penn, Pittsburgh, Pa. T. J. Ess, 1010 Empire Bldg., Pittsburgh, Pa. is managing director.

Sept. 28-30—
National Electronics Conference. Ninth annual conference to be held at the Sherman Hotel, Chicago, Ill. under the sponsorship of the American Institute of Electrical Engineers, the Institute of Radio Engineers, Illinois Institute of Technology, Northwestern University, and the University of Illinois, with participation by Purdue University and the University of Wisconsin. S. R. Collis, N. E. C. Publicity Committee, 208 West Washington St., Chicago 6, Ill., is chairman.

Sept. 30-Oct. 2—
Porcelain Enamel Institute. Twenty-second annual meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. Additional information may be obtained from society headquarters, Dupont Circle Bldg., 1346 Connecticut Ave., Washington, D. C.

Oct. 5-7—
American Society of Mechanical Engineers. Fall meeting to be held at Hotel Sheraton, Rochester, N. Y. C. E. Davies, 29 West 39th St., New York 18, N. Y., is secretary.

Oct. 8-9—
National Conference on Industrial Hydraulics. Ninth annual meeting to be held at Hotel Sheraton, Chicago, Ill. J. G. Duba, Illinois Institute of Technology, Technology Center, Chicago 16, Ill., is secretary.

Oct. 12-13—
First Conference on Mechanisms. Two-day conference to be held at Purdue University, West Lafayette, Ind., under the joint sponsorship of the editors of MACHINE DESIGN and the faculty of the school of mechanical engineering at Purdue University. Additional information may be obtained from the Editor, MACHINE DESIGN, Penton Bldg., Cleveland 13, O.

New Buy-word for Top Value

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Typical design (above) of SHOLD-A-GRIP Bushing and Sprocket with minimum number of teeth.
Typical design (below) of SHOLD-A-GRIP Bushing and Sprocket with maximum number of teeth.

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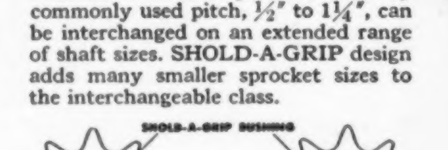
½" to 2½" by 16ths

SHOLD-A-GRIP Sprockets of any commonly used pitch, ½" to 1¼", can be interchanged on an extended range of shaft sizes. SHOLD-A-GRIP design adds many smaller sprocket sizes to the interchangeable class.



Example: Sprocket BKSD, ¾" pitch, 10 teeth, can be used on 9 different shaft sizes, any size from ½" to 1" by 16ths, by inserting the correct size SHOLD-A-GRIP Bushing.

Engineered *originally* and *specifically* for Sprocket drives, SHOLD-A-GRIP Bushings are *not* an "adapted" design. Compare . . . see why SHOLD-A-GRIP means top efficiency, lowest maintenance costs.

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ADHESIVES AND SEALERS

DESIGN ABSTRACTS

Rapidly growing in commercial importance, rubber-base compounds offer a wide range of usefulness in design.

Dr. William J. Clayton — *Technical Director*
and
Roger K. Humke — *Sales Research Engineer*

Adhesives and Coatings Div.
Minnesota Mining and Mfg. Co.
Detroit, Mich.

DURING the last 15 years, adhesives and sealers have gained in popularity and commercial importance as engineering tools at an astonishing rate. Rubber-type compounds in particular have proved to be well suited for use in the transportation industry. Their advantages were first fully utilized by automobile manufacturers. These materials are now used in many places throughout the modern motor car. The related truck and bus industries soon followed suit; the railroad industry then found many applications; and within the last ten years the aircraft industry has used adhesives and sealers on a broad scale.

In general adhesives have the following basic advantages over mechanical fasteners:

1. Adhesives distribute the load evenly over the entire joint area. By giving continuous contact between mating surfaces, they minimize local stress concentrations which are normal with most mechanical fastening devices.
2. Residual elasticity of the adhesive bond, particularly with rubber-base types, serves to absorb some of the stresses created by flexing, vibration, and differences in coefficients of expansion.
3. Weight factor is generally lower, because the weight of adhesive required is usually less than that of the mechanical fastener it replaces. Also use of lighter gage materials is possible.
4. By eliminating voids and gaps, smoother contours can be achieved. Although of some mi-

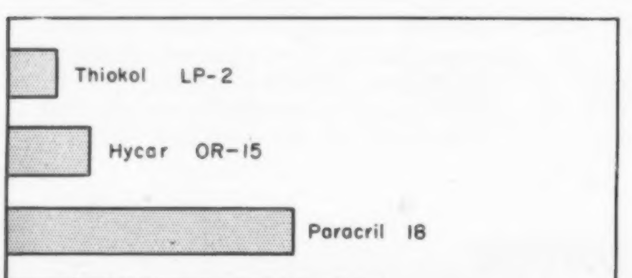

Fig. 1—Per cent swell by volume of synthetic rubbers immersed for 30 days in aromatic (SR-6) fuel maintained at room temperature

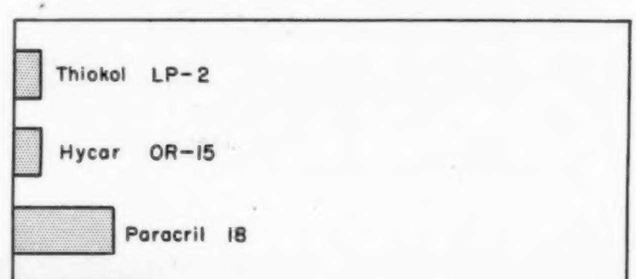

Fig. 2—Per cent swell by volume of synthetic rubbers immersed for 30 days in aliphatic (SR-10) fuel maintained at room temperature

not importance from an aesthetic standpoint, this is of major importance from an aerodynamic or safety standpoint in many applications.

5. Adhesives allow the fabrication of complex shapes which might not be possible with conventional fasteners.
6. Some adhesives have an electrical insulating effect which allows them to function as a barrier to galvanic corrosion between adjoining members of dissimilar metals. In some cases, this property does necessitate special grounding of electrical units to surfaces not insulated by adhesive.
7. An additional corrosion protection factor is often provided by the continuous film of adhesive which prevents passage of corrosive gases, dusts, fluids, and moisture between mated surfaces and into interior voids.
8. In general, adhesives can be applied more rapidly than most fastening devices; a clean surface is the primary preparation required (no drilling, aligning, bucking rivets, etc.). However, it is recognized that allowances in production set-up time must be made for drying and curing intervals.
9. Almost any adhesive compound is capable of application or bonding in a variety of ways, providing flexibility of process or adaptability to other production steps.
10. Where separation of joined parts is desirable for replacement or repairs, application of heat or proper solvent will often allow this to be done without appreciable damage to any of the separate components.

Types and Properties: In the overall adhesive field, the growth in commercial importance of rubber-base adhesives and sealants in particular has been at least as impressive as that of the many other types of adhesives. These materials have unique properties which suit them for numerous special bonding operations. The solvent-release types form strong bonds immediately after assembly and eliminate the necessity of using heat, clamps, or other mechanical or special means of effecting the bond which are always inconvenient and in some cases completely impractical. The rubber-type adhesives, because of their flexibility and nonrigid bonds are resistant to vibration and flexing and can withstand sudden extreme changes in temperature.

Water and Weather-Resistant Adhesives: Types such as those conforming to MIL-C-5092, Type 1, are the simplest ones used by the industry. These products are utilized in bonding felt, fabric, wood, asbestos, and paper to themselves and to aluminum surfaces where there is no requirement for oil or fuel resistance. Adhesives of this group should have the most favorable application characteristics and the easiest cleanup properties. Reclaim rubber and GR-S formulations are qualified under this specification and are broadly used. Both types of materials make use of a relatively inexpensive elastomer and the least expensive adhesive solvents, such as naphtha or similar petroleum fractions. They are easy to clean up and fall in the relatively inexpensive price class of $2 to $3 per gallon. They have good resistance to water and generally adequate weathering characteristics.

Oil-Resistant Adhesives: Bonding of neoprene tubing, sheeting, foam, and gaskets to themselves and to vinyl is usually accomplished by neoprene adhesives like those conforming to MIL-C-5092, Type 2. In this case, since high oil resistance is required and specific adhesion to neoprene is essential, the reclaim rubber and GR-S type adhesives are entirely unsuitable. The application of neoprene adhesives is somewhat more difficult than that of the first class of materials discussed but is considerably easier than the application of other synthetic adhesives. Neoprene as an adhesive base requires the use of xylene or related aromatic hydrocarbon solvents, which are more expensive than naphtha but far cheaper than the esters and ketones required by other synthetics. Due to their relative ease of application and price of $4 to $5 per gallon, neoprene adhesives are widely used in the automotive and aircraft industries. The principal handicap of neoprene adhesives is their tendency to gel while standing, but intensive effort is being devoted to correcting this deficiency.

Aromatic Fuel-Resistant Adhesives: Products like those conforming to specification MIL-C-5092, Type 3, must be suitable for use in bonding nitrile rubber tubing, foam, sheeting, and gaskets to themselves and metal and in areas which come in direct contact with aromatic fuels. These adhesives also must withstand the plasticizers in nitrile or vinyl stocks and thus give permanent bonds between highly plasticized materials. None of the presently known elastomers meet these requirements except the nitrile rubbers. These Buna-N materials are expensive and are difficult to process. They require the relatively expensive esters or ketones as solvents, and as a result, sell for about $5 to $6 per gallon, have a short bonding range and are relatively difficult to use. They are, however, broadly used for a number of operations in which other adhesive types are entirely unsatisfactory.

Aliphatic Fuel-Resistant Adhesives: The Air Force specification MIL-C-4003 requires an ad-

Table 1—Comparison of Adhesive Rubbers

Rubber Type	Price* (cents/lb)	Specific Gravity	Resiliency	Tensile Strength	Ultimate Elongation	Natural Aging	High Temp. Aging†	Aliphatic Resistance	Best Solvent§
Natural	60	0.93	Best	Best	Best	Average	Best	Poor	Petroleum
Neoprene	45	1.25	Good	Good	Good	Good	Average	Average	Aromatic
GR-S	24	0.94	Average	Fair	Fair	Fair	Fair	Poor	Petroleum
Nitrile (Buna-N)	50	1.00	Average	Average	Fair	Fair	Fair	Good	Ketone
Polysulfide (Thiokel)	100	1.27	Poor	Poor	Poor	Good	Poor	Best	Chlorinated Hydrocarbon

*Approximate market prices per dry pound in December 1952. †Resistance to aging at 200 to 300 F temperatures. §Solvents rated "best" considering both solvency and cost.

hesive which meets almost all the performance requirements of the previous three types. The nitrile rubber formulations which comply with this specification have slightly more favorable application characteristics than the products referred to in the previous classifications, but they are still far more difficult to apply than reclaim rubber and GR-S formulations, or neoprene. Materials conforming to this specification are the nearest general-purpose adhesives known to the industry. They have the limitation of not being resistant to continuous immersion in aromatic fuels, but their resistance to aliphatic types is excellent. These adhesives can be used for almost any aircraft bonding application except high-strength metal-to-metal bonds, and they are therefore used in most repair work. In new plane construction, the industry seems to prefer the less expensive, easier-to-apply adhesives in all locations where they are suitable.

SPECIAL-PURPOSE ADHESIVES: Each class of adhesive described previously has its own particular merits and weaknesses; but each of them can be successfully used for a wide variety of applications, and hence they are usually referred to as general-purpose adhesives.

There are a few materials which are so highly specialized that they have a very limited usage. For example, the adhesive required under Air Force specification No. 26610 must bond cured nitrile rubber patches to cured nitrile rubber liners in both self-sealing and collapsible fuel cells; this bond must not be deteriorated even after several years of constant immersion in aromatic fuels. The resin-base adhesive which is widely used for this application has been exceptionally successful in fuel cell repair, but up to the present time has never been used on a major scale for any other application.

RUBBER-BASE SEALERS: Perhaps the most important and difficult requirements for rubber-base sealers are those encountered in the sealing of integral fuel tanks. The most broadly used sealants of this kind, complying with specification MIL-S-7502, are those based on Thiokol liquid polymers. These compounded viscous forms of 100 per cent Thiokol polymers are catalyzed by the addition of proper activators and then used in the 4 to 5-hour working time before the sealant sets up. In 24 to 48 hours these products cure to a rubbery mass which has exceptionally good resistance to low-temperature vibration and exposure to aromatic fuel. These sealants are also outstanding because they relax under pressure, and therefore remain in place even after the metal members which they seal have been disturbed.

Water-dispersed pastes of Thiokol latex base are also used for integral tank sealing in aircraft which are not subject to extremely low temperatures or extremely high altitudes. The principal advantage of these water-type putties over liquid polymer or LP-2 types is in their ease of application.

SPECIAL-PURPOSE SEALERS: Like adhesives, there are some requirements for sealants in aircraft which require special-purpose materials. An example is the so-called fire wall sealant, which is used to prevent flash fires from spreading beyond the engine compartment and which must resist temperatures up to 2000 F for brief periods of time. Fortunately, it is not necessary for this product to retain its rubbery properties after exposure to this temperature. It is therefore possible to use an inorganic base which is bound together by an elastomer. Upon heating, the elastomer is burned away and the inorganic material is fused to a continuous mass.

POTTING COMPOUNDS: Experience has demonstrated the absolute need for potting compounds in certain types of electrical connections. Without proper protection some electrical connections will corrode

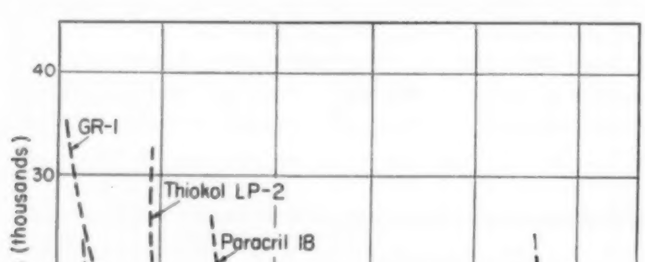

Fig. 3—Low temperature properties for various rubbers measured in terms of absolute modulus

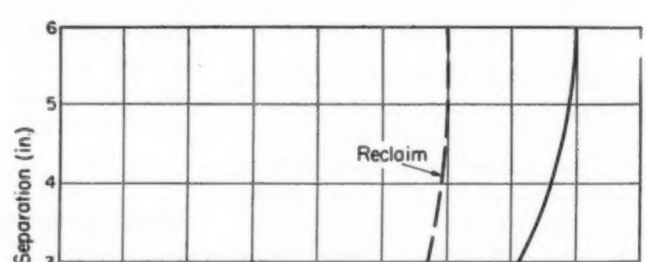

Fig. 4—Heat softening properties for four types of adhesives

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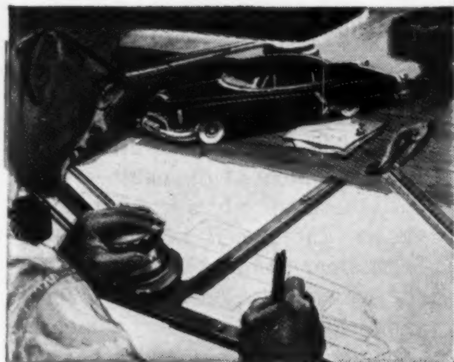

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Design Abstracts

to the point where the current is cut off and the equipment will not operate properly. It seems likely that within the next year or two the aircraft industry will use potting compounds on a very much greater scale than they have been used in the past. The most promising materials so far developed for this application are of a Thiokol LP-2 type.

Compounding: Adhesive manufacturers are often accused of putting gold dust in their aircraft adhesives. Most of the engineers who use these adhesives understand the basic differences in the various elastomer compounds, typical uses of which have been given. These engineers also know something about the progress which the rubber industry has made through compounding. They therefore quite justifiably wonder why the adhesive manufacturers do not learn to compound from neoprene or nitrile rubbers products which have the properties of the Thiokols; or why the manufacturers do not make adhesives from GR-S which would be adequate replacements for the more expensive neoprene products.

It is true that compounding can accomplish veritable miracles with a given rubber polymer. The first classical example of this was the discovery of sulfur vulcanization, which transformed natural rubber from an interesting plaything with little practical value into the remarkable tool of industry which it constitutes today. The use of carbon black as a reinforcing pigment was the second milestone in the evolution of rubber compounding techniques. This compounding discovery increased the abrasion resistance of natural rubber manyfold and made possible the modern automobile and truck tires.

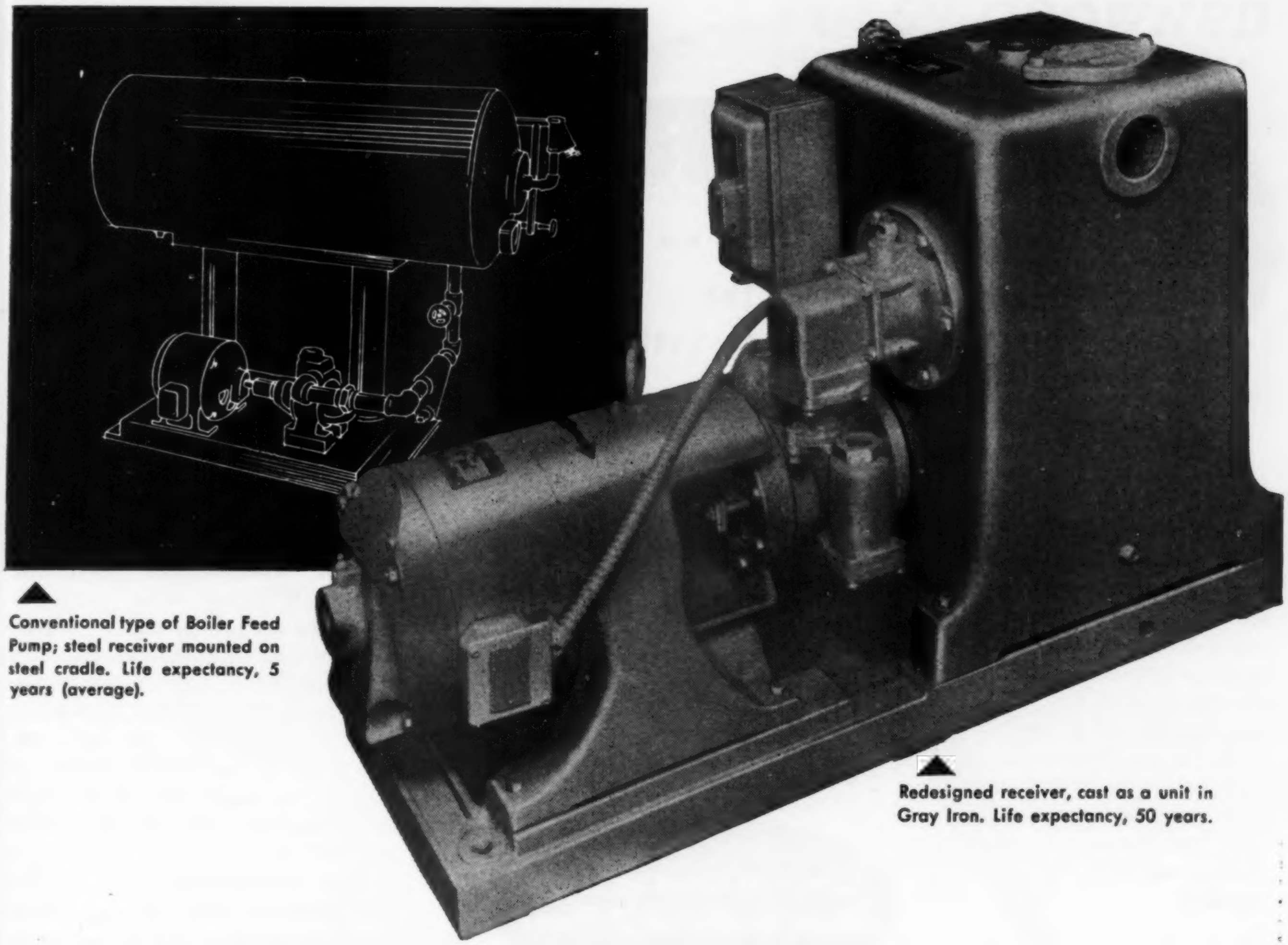
Compounding of rubber has done much more toward enhancing physical properties than it has toward changing chemical properties. For many years hundreds of capable rubber chemists tried in vain to learn how to compound a natural rubber stock with oil-resistant properties. In this fruitless quest they could be likened to the al-

chemists of old, who tried to transform lead into gold. Even after neoprene, Thiokol and nitrile rubbers became available commercially, efforts were made to replace these specialty rubbers with inexpensive natural and synthetic rubbers. Since no appreciable success has come from all this effort, the rubber and adhesive compounder has learned to place emphasis on the selection of the proper elastomeric base as well as the use of the proper modifying agent.

In TABLE 1 a comparison is made of rubbers used in adhesives and a few additional qualitative differences between these elastomers are pointed out. Specific values of physical properties vary considerably with each separate grade and compounding.

Fuel Resistance: To illustrate the comparative fuel resistance of the synthetic rubbers listed in TABLE 1 on a quantitative basis, consider the graphical presentation of the per cent swell while in SR-6 and SR-10 fuels, as shown in *Figs.* 1 and 2. The materials were immersed in the fuels for 30 days and the fuels were maintained at room temperature. These graphs demonstrate the unusually low swell and, therefore, excellent fuel resistance of Thiokol and nitrile rubbers. It will be observed that the advantage of Thiokol over nitrile rubber is more apparent in the SR-6 fuel, which contains 30 per cent aromatic hydrocarbon, than in the SR-10 fuel, which is 100 per cent aliphatic fuel.

Low Temperature Properties: In view of the importance of low temperature performance of adhesives and sealers, the information in *Fig.* 3 is most pertinent. One of the most satisfactory ways of measuring low temperature properties of rubbers has been found to be in terms of modulus. This represents a measurement of strain resulting from an applied stress or the distortion resulting from an applied load. To obtain the data for this graph. dumbbell shaped samples of the elastomer were subjected to an angular torque with the load and angle of deflection



Conventional type of Boiler Feed Pump; steel receiver mounted on steel cradle. Life expectancy, 5 years (average).

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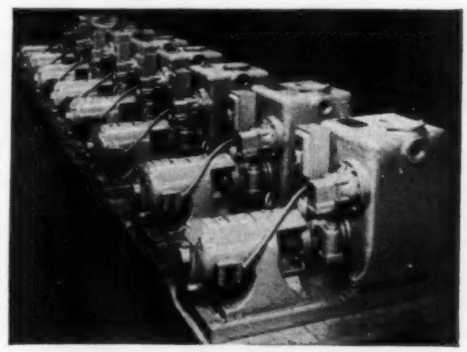

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measured at the various temperatures.

The low temperature data in *Fig.* 3 is based on comparisons of gum stocks. Improvement in low temperature flexibility of nitrile rubbers can be obtained by the use of proper plasticizers. Under the conditions of this test, Thiokol retains its flexibility to approximately −65 F, paracrill to −50 F, and Hycar OR −15 to −5 F. Under other test conditions, the brittle points might be found to be much lower but always in the same relative order.

High Temperature Softening Points: The softening point of adhesives is particularly important in the automobile industry and somewhat important in the aircraft industry. The curves in *Fig.* 4 are based on data obtained from peel-back bonds of canvas to steel for four types of adhesives. The samples were allowed to stand for 15 minutes at each temperature, with a load of 100 grams per inch of bond, before measuring the length of peel back on slippage. It will be observed that the reclaim bond began to slip at 155 F and soften markedly at 220 F. The GR-S bond slipped at 175 F and softened at 240 F. The neoprene bond slipped a little at 260 F but cured so rapidly at that temperature that it did not slip further even at 310 F. The nitrile rubber bond cured before slipping and hence was unchanged through 310 F. The cured neoprene and nitrile bonds did not soften further, but at 375 to 400 F they failed by charring.

Reinforcing Pigments and Fillers: Contrary to popular belief fillers are not used solely to reduce cost. They contribute many valuable properties. They often improve physical strength as shown by use of carbon black in automobile tires to improve abrasion resistance and in synthetic rubbers to improve fuel resistance. Fillers may be necessary to reduce the tackiness of the binder and allow greater ease of handling of precoated surfaces. Fibrous fillers such as asbestos are of value where

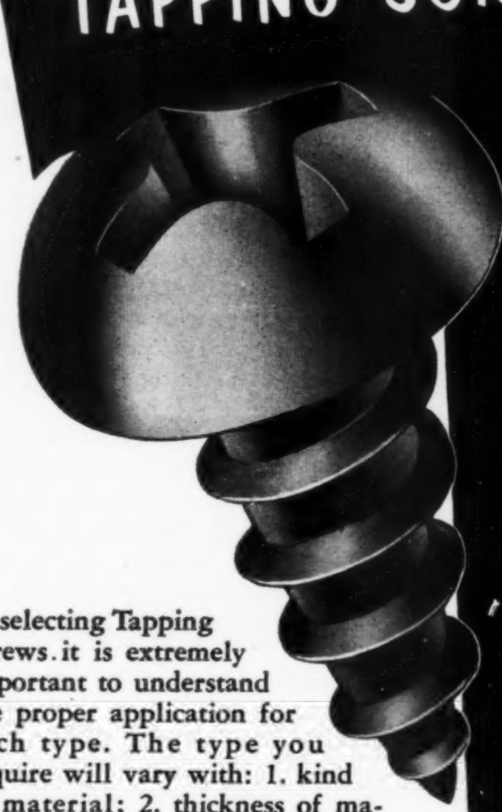
application of greater thicknesses of material is desired. In many instances, the drying rate of an adhesive may be increased by the addition of proper fillers.

Modifiers: For simplification "modifiers" refer to all the various materials that are used to change the properties of the binders. These include the following:

1. Accelerators and other cross-linking or polymerizing materials are used with rubber and synthetic rubber. This category refers to the use of sulfur in natural rubber, reclaim rubber, and some synthetic rubber compounds along with the use of peroxides in the curing of Thiokols.
2. Plasticizers are the materials intended primarily to decrease brittleness; improve resiliency; and sometimes improve adhesion of the compound by enabling better wetting of surfaces. Usually these ingredients improve the low-temperature flexibility, but at the same time adversely affect the high temperature of fuel resistance of the binder. Plasticizing agents can be divided into two categories—mechanical and chemical. The mechanical type includes the elastomeric materials, which modify the binder simply by their physical presence. The chemical type of plasticizer is in effect the reverse of the curing or vulcanizing agent in that it usually functions by causing depolymerization or by reducing the cross-linking of the binder. The latter type is not employed as extensively as the mechanical type of plasticizer.
3. Tackifying agents include those natural or synthetic resins which are added, where necessary, to improve the adhesion of the binder to various types of surfaces.
4. Stabilizing agents are chemicals added to improve the resistance of the base material to heat, light, or chemical breakdown.

Vehicles and Solvents: A few rubber-like adhesives or sealers contain no volatile component such as the Thiokol LP-2 sealers. The majority of adhesives, coatings and sealants contain from 40 to 80 per cent solvent by volume. These solvents are necessary as the vehicles

TAPPING SCREWS



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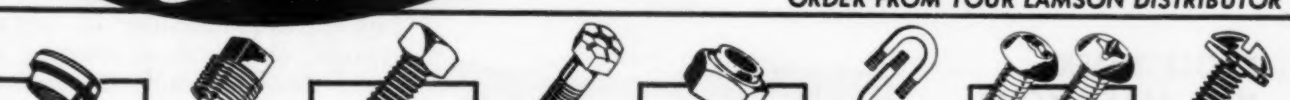

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which put the basic materials in a form which allows easy application, and then they are volatilized to deposit a uniform film which performs the intended job. The proper solvent for any given compound is one which is capable of dissolving or dispersing the other ingredients, allows convenient application and volatilizes rapidly enough to be practical for the production or assembly operation involved. Since some of these properties are mutually exclusive, it becomes necessary to compromise between the advantages of a high boiling solvent with a long working range and a low boiling solvent which causes the adhesive to dry rapidly.

Obviously volatility of the solvent is not the only limiting factor, since solubility of the binder enters into the picture. For natural rubber, reclaim rubber, and some synthetics, petroleum or naphtha-type solvents are widely used. For neoprene compounds, aromatic hydrocarbons are required, for the Thiokol and nitrile types it is necessary to use ketones, esters, or chlorinated solvents. It is an unfortunate circumstance that the most expensive binders in general also require the most expensive solvents, thus preventing balancing or offsetting of cost.

It is sometimes possible to use a dispersed elastomer, such as Thiokol latex, or to disperse the basic ingredient in water. This has some advantages from a cost and ease-of-handling standpoint. The principal handicaps of water dispersions in general are in their tendency to corrode metal surfaces on which they are applied, and their inability to withstand repeated freezings and thawings.

Future Possibilities: Constant effort is being directed toward improvement of present standard compounds in their convenience of application, durability of performance, and relative economy of use. Quite naturally, the bulk of such effort is concentrated on the modification of present commercially available elastomers.

Use of adhesion-promoting additives in elastomers is an essential

approach toward improving the performance of both adhesives and sealants. Isocyanates are being investigated for this purpose because of their apparent beneficial effect on adhesion of elastomer compounds to both fabric and metal materials.

The use of epoxy resins themselves is well-known, but certain combinations of them with elastomers and/or other resins show definite promise in the formulation of improved potting compounds, fuel tank sealants, and related products.

Silicone rubbers have also been known to the aircraft designer for quite some time. However, certain forms of them now available to the sealant compounder offer definite advantages in the extremely wide temperature performance range which they afford. Although these materials still lack resistance to aromatic fuel, much has been accomplished in making them resistant to oil and aliphatic fuels.

From a paper entitled "Rubber-like Adhesives and Sealants" presented at the Seminar on Adhesives and Sealants sponsored by the Southern California Section of SAE in Los Angeles, Calif., January 1953.

Fundamentals of Bimetal Performance

By C. F. Alban
Chief Engineer
W. M. Chace Co.
Detroit, Mich.

THERMOSTATIC bimetal is composed of two or more lamina of metallic alloys having different coefficients of expansion and physical properties. Physical properties sought determine which alloys are to be used in combination. Properties usually considered are coefficient of expansion, modulus of elasticity, elastic limit after cold rolling, electrical conductivity, ductility, metallurgical stability, and strength at various temperatures.

Each thermostatic bimetal must have a low and a high expanding lamina. Properties of the combina-

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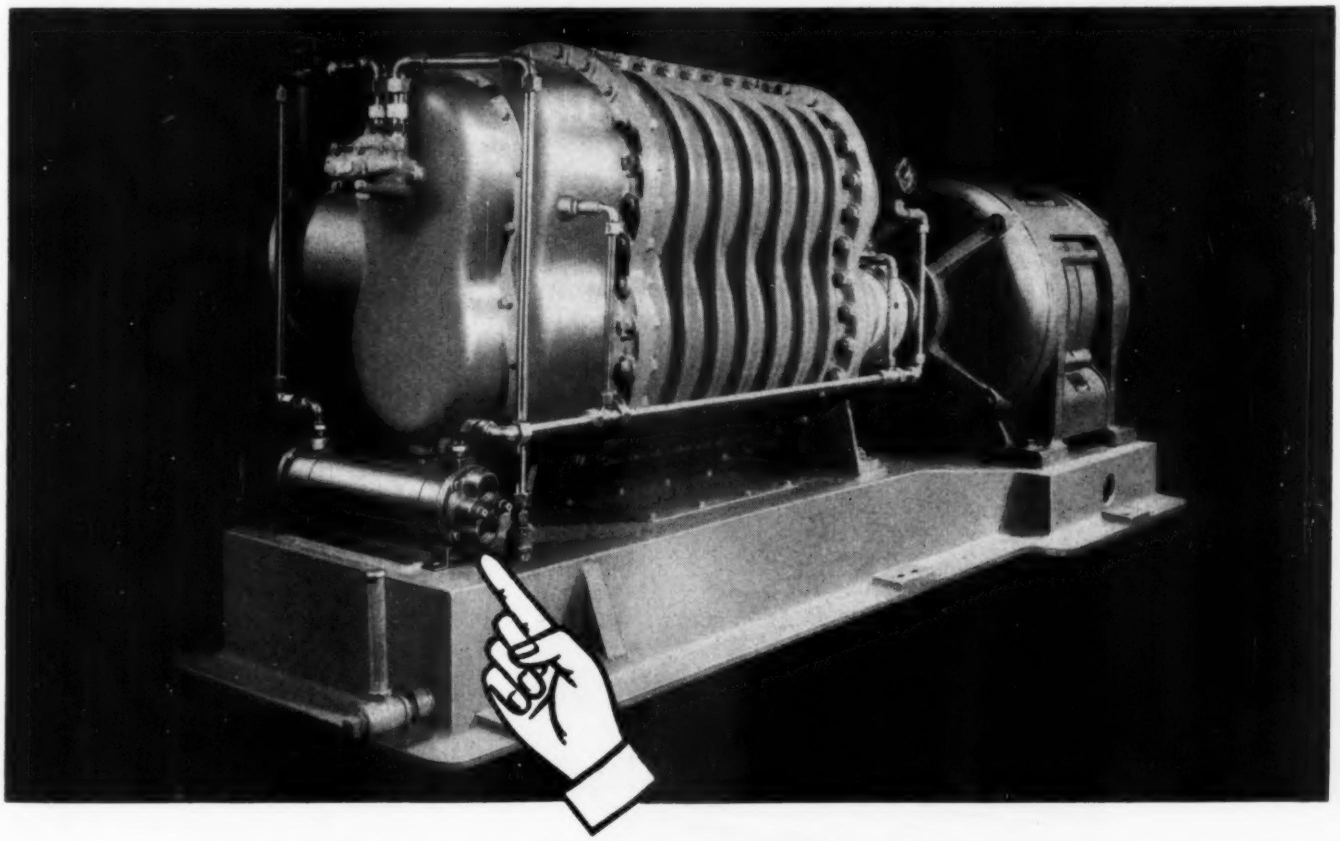

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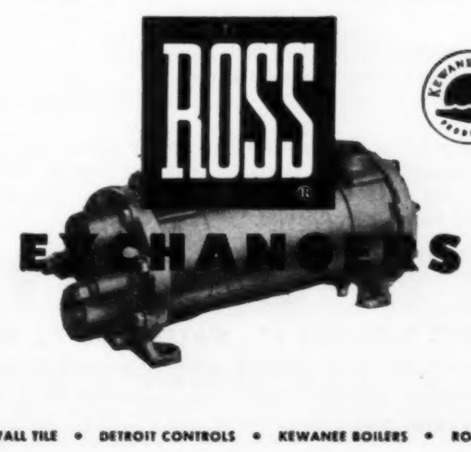

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tion may be modified by the insertion of a third layer in the center. This is usually done to obtain the electrical characteristics desired.

The fundamental property of all thermostatic bimetals is the ability to change curvature with temperature change. With change in curvature a force is developed. This force is the same as that developed when a beam is bent or curved an equal distance.

Thus, thermostatic bimetal is a simple, rugged method of changing heat into mechanical work. Mechanical work developed with temperature change is the basis of many modern controls or devices.

Composition: Low expanding laminae are generally selected from nickel-iron alloys of 36 to 50 per cent nickel. Lowest coefficient of expansion occurs with 36 per cent nickel-iron, which will provide the highest deflection rate for a given thermostatic bimetal for temperatures up to approximately 350 F. This temperature called the inflection temperature is raised with increasing nickel content. However, maximum deflection rate is sacrificed for higher inflection temperature. For example, with a 50 per cent nickel-iron low-expanding lamina, practically straight line deflection is obtained up to 800 F.

High-expanding laminae are generally selected from the nickel-chromium-iron series. The best all-purpose bimetals use these alloys. They have useful deflection properties up to approximately 800 F. However, in recent years an alloy of 72 per cent manganese, 18 per cent copper, and 10 per cent nickel has come into prominence, but its use is limited to temperatures up to 400 F. This alloy has an unusually high expansion coefficient and makes possible a thermostatic bimetal with an exceptionally high deflection rate.

Bonding: Some unusual metallurgical problems have been presented by the bonding or laminating of the various alloys into an integral sheet. It is desired to utilize all of the physical properties to the best advantage, which rules out any methods using brazing or

soldering. The various laminae are joined together by diffusion welding. This is welding in the solid state without the use of any intermediate material. In this process, temperature, pressure atmosphere, and surface preparation are the important factors.

Maximum Safe Stresses: When the combination of thermal and mechanical stresses exceeds the elastic limit of the thermostatic bimetal element, a permanent set occurs. This results in a change in calibration of the thermostat in which the element is used and is generally progressive if the unfavorable conditions are continued. It must also be recognized that there are other factors which may cause the thermostat to change its calibration. Among these factors are weak mounting, relieving of mechanical stresses in associated structural members, excessive restraint of thermal deflection, shock, corrosive conditions, changes in the electrical contacts, and stresses set up in mounting the thermostat.

Long term tests have been run below the elastic limit to determine the effect of time. Any precipitation hardening effects must not be present in bimetal components. A gain in hardness on a durability test usually is indicative of these effects. Dimensional changes usually accompany hardening of this type and entail a change in calibration of the thermostat in which this type of bimetal was used. Also on durability tests there should not be an appreciable drop in hardness.

Applications: In the past there have been applications where thermostatic bimetal was misapplied. Stresses and temperatures have been too high, corrosion and erosion have been problems and poorly constructed devices with inferior materials have caused troubles.

The thermostatic bimetal element is usually blamed for all troubles. The fact that a loose, weak mounting was used is sometimes overlooked. All components should receive careful attention from the designer. Careful selection of all

(*Continued on Page* 282)

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(*Continued from page 279*)

materials and their assembly is paramount.

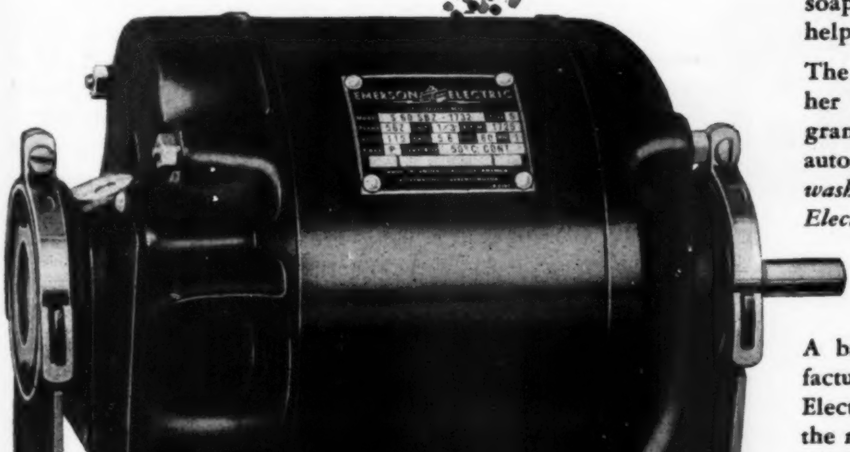
When bimetal thermostats change calibration with temperature and time there is always a reason. Sometimes careful analysis of each component is required to locate the source of trouble. For example, consider a well constructed thermostat of the finest materials available. Occasional changes in calibration occur in shipping during the winter. Examination shows that the bimetal is under mechanical restraint to obtain the required operating temperature. In this case additional stress is built up in the bimetal due to being transported through a cold area. Additional stress built up is sufficient to exceed the elastic limit in the bimetal, and a change occurs. This could be baffling, as the thermostat was all right when it left the factory. Laboratory tests at the destination show the calibration is off for no apparent reason. This problem can be solved by either redesigning the thermostat or using a lower deflection rate bimetal.

Design Techniques

Usually, the best method is to design a thermostat where the bimetal is free to deflect until the control temperature is reached. Another method is to limit the stresses within the elastic limit of the bimetal. The most practical way of lowering stress is the selection of a bimetal with a lower deflection rate. If initial thermal restraint or mechanical loading is present in the thermostat, this value will be less. The designer has to determine what these stresses are and allow for them. One severe test is to place the thermostat in dry ice over night. If no change in calibration occurs, the thermostat should be satisfactory from this stress standpoint.

Another source of trouble encountered in thermostatic devices using bimetal is friction. Latch surfaces should be hard and polished. All other components should be made with a precision in keeping with the quality of the device. Some devices have shafts that

clothes come clean ... at home



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seize when heated or cooled. In these cases, the bimetal pulls itself up by its "boot straps" and a set occurs.

Another testing technique is to operate the thermostat over a long period of time at the maximum temperature of use. Any change from any cause will show up as a decrease or increase in the temperature controlled. In the overstressed parts this change can show up within an hour. Defective mountings and dimensional changes usually take longer.

Modern thermostatic bimetals are sturdy and rugged materials. They can be formed, cut, punched, embossed, riveted, spot welded, silver brazed, or soldered. They have high elastic limits imparted by cold rolling and can do mechanical work proportional to the square of the temperature change. Temperature change can be produced by heat from radiation, convection, conduction or electric current.

From a paper of the same title presented at the AIEE Appliance Technical Conference in Louisville, Ky., April 1953.

Limitations of Contour Forming

By Dr. George Sachs
*Director of Metallurgical Research
Syracuse University
Syracuse, N. Y.*

CONTOUR forming can be accomplished in a number of different ways. As used here the term refers to the curving of both sheet metal sections and rolled, or extruded, shapes. Basically, this involves two distinctly different methods, either bending or stretching.

This discussion is primarily devoted to the limitations of bending-type contour forming. The rather lengthy term is used to emphasize that contour forming of a thin-walled section is governed by rules which are more complex and consequently different from those which govern the bending of a solid, rectangular or circular cross-section. The bending of rectangular sec-

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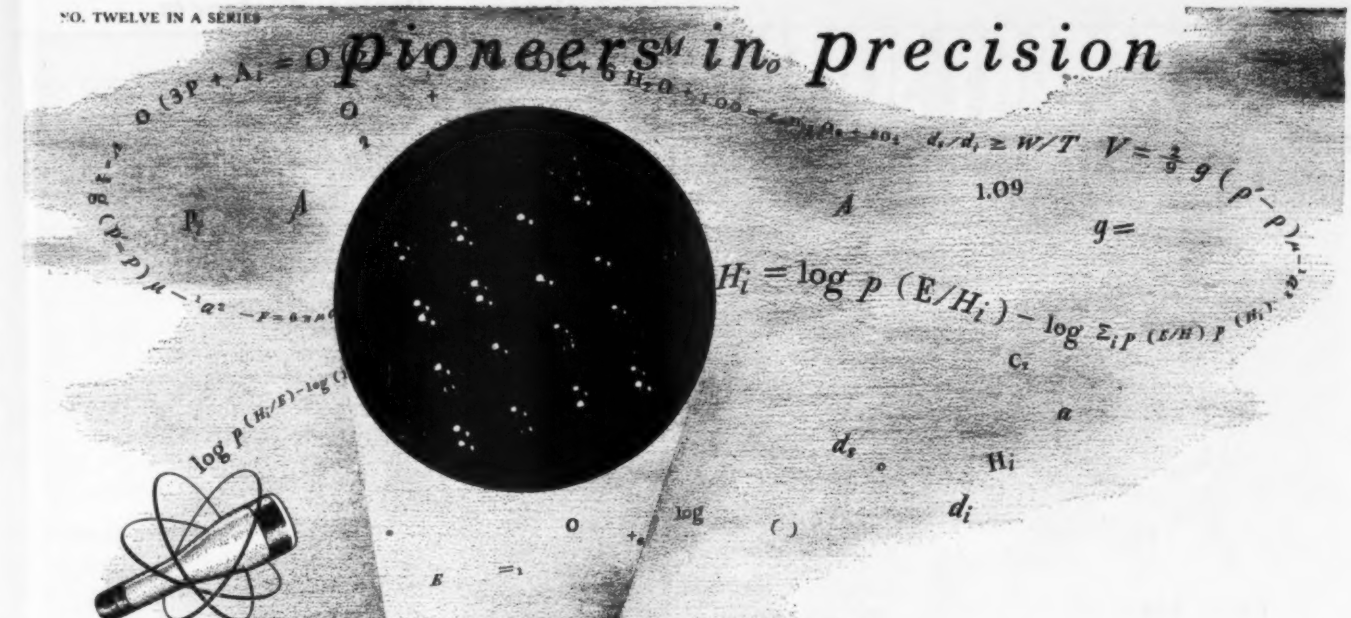

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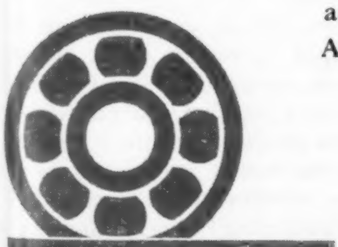

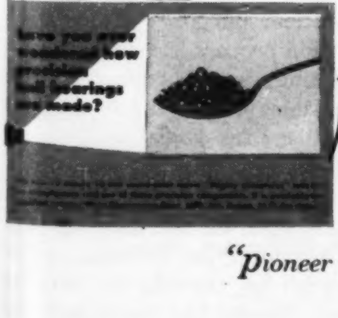

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tions also comprises a border case of contour forming.

Process Limitations: In general three types of failures may limit the process of contour forming and the usability of the formed part. These are:

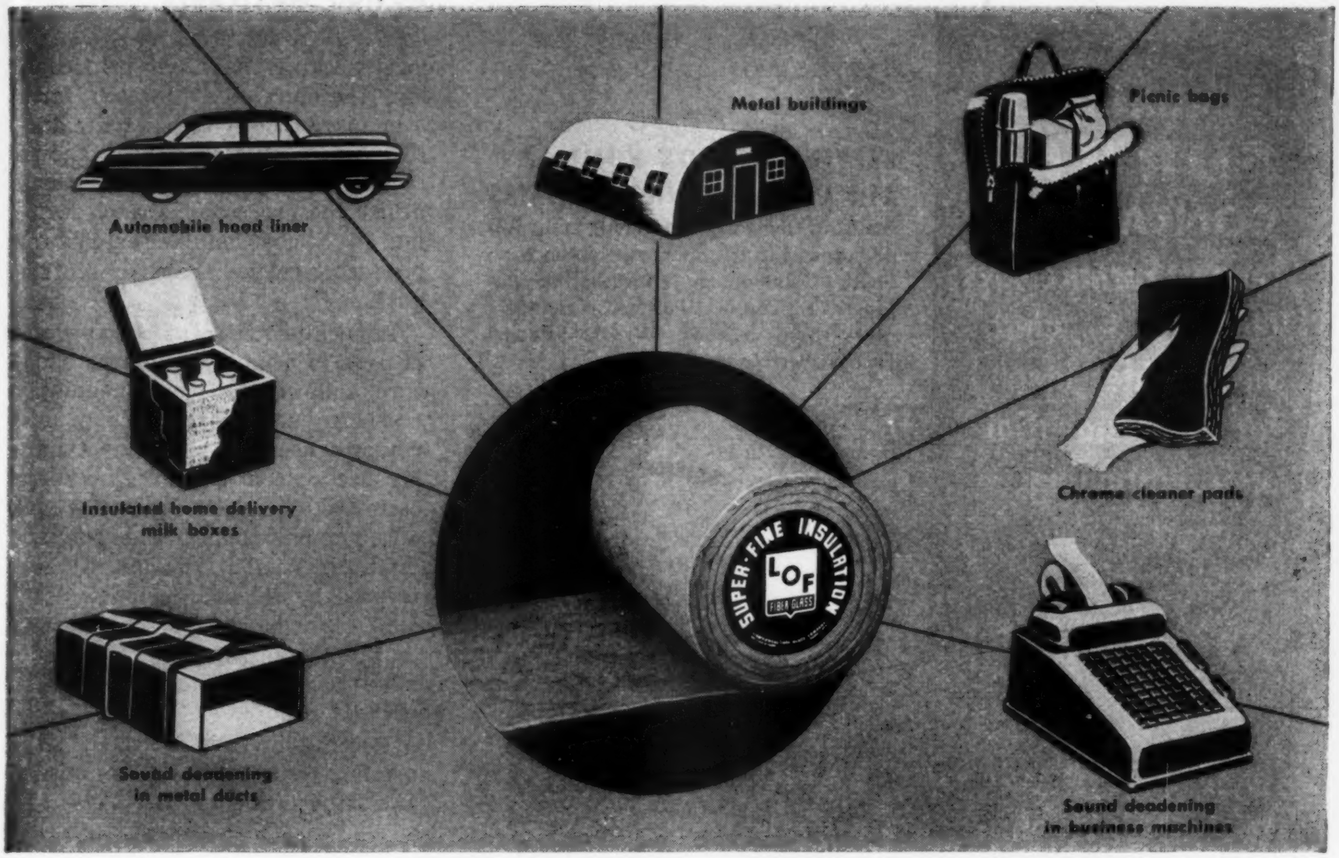
1. Metal cracking at some location on the outside or convex surface.
2. Metal wrinkling at the inside surface or part collapse.
3. Failure to conform to the desired shape because of springback.

While all three failure types are of concern to the practical engineer, wrinkling, collapsing, and springback can be controlled to a considerable extent by constructional and operational measures. The limitations imposed by cracking depend entirely upon two factors which are frequently outside of the control of the shop: (1) the properties of the material and (2) the shape of the part.

Cracking of a metal is usually caused by tension stresses acting, as a rule, in the direction perpendicular to that of the developing crack. For bending and contour forming the inside surface of the metal is subjected to tensions and stretching.

Stretching of any material leads to failure if the stretch reaches certain limits. It has been recognized that this failure may be one of two different types, either fracturing or necking. In a conventional tensile test on a ductile material, the originally prismatic or cylindrical length of the specimen first stretches rather uniformly under load until a certain limit called "uniform elongation" is reached. On further loading, the stretching becomes localized within a narrow range, the "neck", where the metal may be stretched much further before "fracturing" occurs. However, for practical purposes of forming, the tension process is terminated by the necking. In a tension test it is not possible to retard the necking and utilize the total stretching ability or "ductility" of the metal.

As a consequence, ductility has little influence on the forming limit of tension-type forming processes such as curving by stretch forming. Hereby the metal fails by



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Almost every major industry finds profitable use for Super·Fine. It is used for insulation in aircraft, automobiles, railroad cars, ships, refrigerators, air conditioners, ducts, steel buildings and many others.

But there's more. Super·Fine has been used in profitable new ways such as chicken nests, blackboard erasers, protective wrapping and chrome cleaners. Other uses are being discovered daily.

Put your imagination to work. Perhaps L·O·F Super·Fine Fiber·Glass is the magic material you've been seeking. Contact your nearest L·O·F office (offices in 26 major cities). Or write Libbey · Owens · Ford Glass Company, Fiber · Glass Division, 3683 Wayne Building, Toledo 3, Ohio.

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Design Abstracts

local necking. A rather brittle alloy, therefore, may stretch form even more readily than a quite ductile metal. This is a well known fact in the aircraft industries which utilize the process of stretch forming to a considerable extent for producing curved shapes and other parts.

In such processes as bending, the process is usually limited by fracture following a correspondingly extensive uniform stretching. The performance of a simple rectangular section on bending has been stated to comprise a borderline case of contour forming. The forming limit in bending is usually expressed by the minimum bend radius, R, or preferably by its ratio to the thickness, R/t, to which a rectangular section can be bent without failure.

Recent investigations of the bending process have established two important facts:

1. The bend radius depends upon the cross-sectional dimensions of a rectangular section.
2. This relation, i.e., the effect of the width-to-thickness ratio on the bend radius, is very different for different materials, and also for different tempers of the same metal.

An annealed and ductile metal, such as aluminum alloy 75S immediately after quenching (from the heat treating temperature), will allow a very wide part to be bent only to a radius that is considerably larger than that for an approximately square section. This wide section can be also bent much further than a very narrow section. For a hard and comparatively brittle alloy, such as the fully hardened condition, 75S-T, of this aluminum alloy, the differences are greatly reduced. Its bend radius is much larger than for the more ductile alloy if the section is wide. However, if the section is very narrow, both alloys approach nearly identical bend radii.

The wider the part, the less free it is to shrink or stretch transversely and thus to develop a saddleback contour. As a consequence, tensions develop at the outer surface not only in the longitudinal but also in the transverse direction. Such a "biaxiality" reduces

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ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921

the ductility of any metal. A systematic investigation of the bending process has shown that the transverse tensions and the biaxiality increase with the width of the section and that the ductility correspondingly decreases.

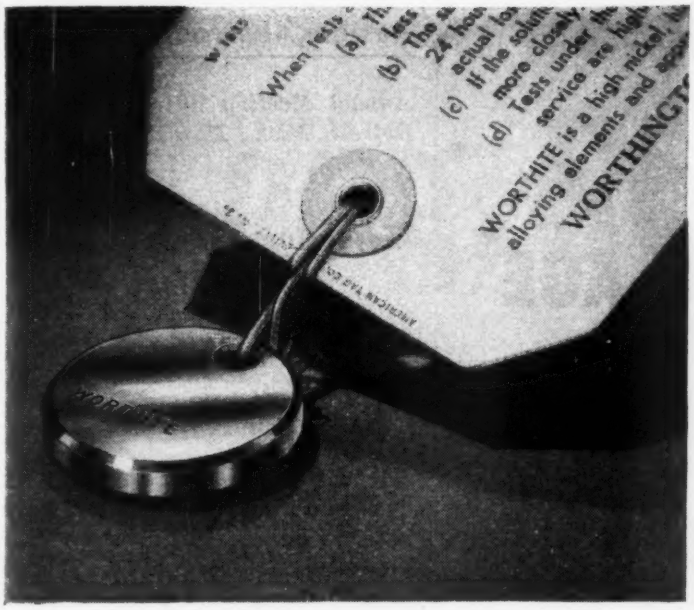
Very narrow sections are found to fail by necking and not by rupturing. The behavior of narrow sections on curving is decisive for the forming limit of many contour-formed parts. The bending of such narrow sections has shown that the narrower the section, the lower the necking strain which then comprises the forming limit, while with increasing width the necking limit approaches the same value as the rupturing limit.

More Data Needed

On the basis of the foregoing it now becomes possible to evaluate the performance of contour-formed sections. At present, the data available is very meager and not sufficient to predict the forming limit of any particular section in a specific metal. The complexity of the problem is explained partly by the fact that rupturing and necking of two different portions of the section may compete with each other. In addition, the neutral plane of a section may be positioned closer to one surface than to another, and the surface strain at the outer surface decreases as the neutral plane approaches this surface. Finally the forming limit for either fracturing or necking depends upon the strain gradient, i.e., the rate of change from stretching to shrinking over the height of the part.

As a consequence of these complex and conflicting influences, the radius of curvature for a given material varies greatly, not only for different section shapes but also for any given section shape, depending upon the relative dimensions of its components. Considering the large number of variables involved in this problem, it is clear that only very extensive systematic investigations will be able to supply satisfactory design data for a range of cross-sectional contours.

From a paper entitled "Some Basic Relations in Contour Forming" presented at the Twenty-First



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The wrong pumps can impair your product's efficiency; the right one can greatly enhance it. To aid in this selection, Worthington offers complete lines of every major type pump. Here are three examples of the way Worthington design overcomes specific pumping problems.

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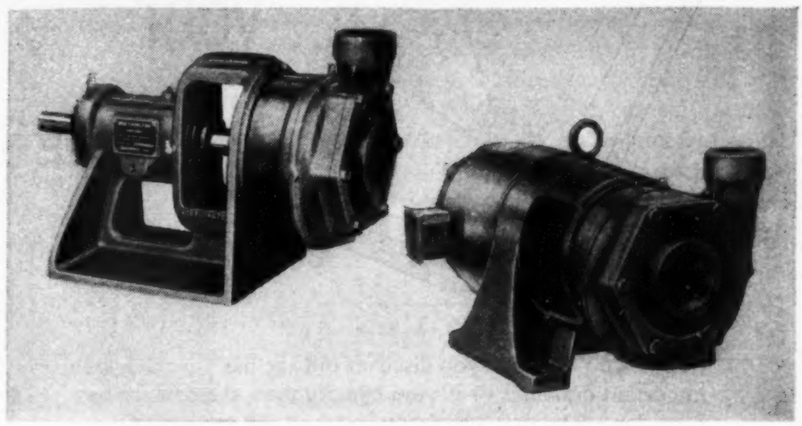

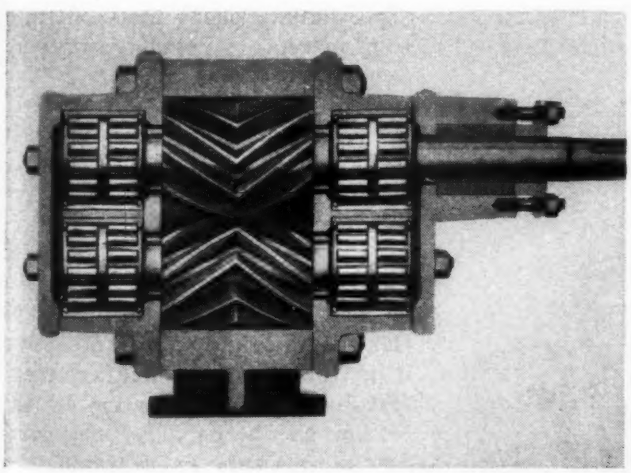

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For further information on Centrifugal, Rotary or Worthite pumps, write to Worthington Corporation, Pump and Compressor Merchandising Division, Section PC.3.13, Harrison, New Jersey.

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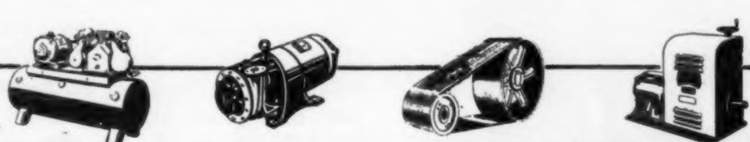

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Design Abstracts

Annual Meeting of the ASTE in Detroit, Mich., March 1953.

Selecting and Specifying Materials

By J. W. Frazier
Materials Engineering
Hughes Aircraft Co.
Culver City, Calif.

WHEN to select and specify materials could well have been a subject for debate thirty years ago. Selection could have been made at almost any stage of producing an item. Lack of concern or lack of knowledge or both, contributed to disorganization of effort on materials problems. Today this is changed, but there still remain three stages during which materials can be selected.

1. Design (including preliminary and product)
2. Production Engineering
3. Manufacturing

Since manufacturing is guided by engineering requirements and specifications, it is generally conceded that the selection of materials should not be made at this point except in cases of extreme necessity.

There is a general feeling that the need for materials begins with the product engineer. This could very well be true when his activity includes product design, which it does in the majority of cases. However, product design should in no way be confused with production engineering whose objective it is to plan for manufacturing. Material problems relative to selection should only be incidental by the time they reach production engineering.

The increasing importance of complete design is becoming more pronounced every day. Specifications governing models, prototypes and design itself are tightening to the extent that machines are expected to be designed as functionally and physically potential production items. This is not an uncommon requirement in many industrial and government specifications and it will probably become

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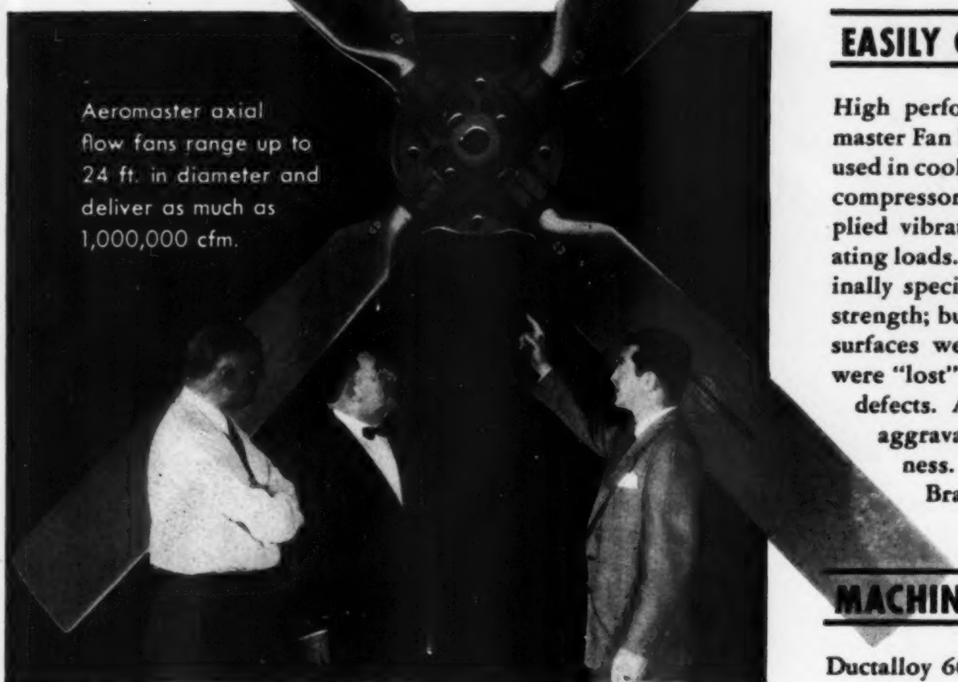

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Ductalloy 60 is a ferritic iron having 60,000 psi minimum tensile strength, 40,000 psi minimum yield strength and 10% minimum total elongation, all guaranteed. Use of this material has virtually eliminated the high scrap loss formerly encountered when defects were uncovered during machining of castings made in either steel or malleable iron.

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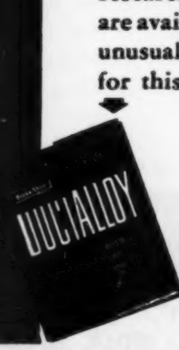
your problem — Ductalloy may solve your problem if it involves economical production of complex metal shapes that are difficult to cast in steel, expensive to forge or lacking strength in gray iron. Brake Shoe's experience, research laboratory and experimental foundry are available to help you best utilize Ductalloy's unusual combination of characteristics. Write for this new technical bulletin today.



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more common. If this be true, it is not unreasonable to assume that the basic selection and specification of materials should be made in the design stage of a machine or even in the preliminary stage of design. Certainly, there is every opportunity for the materials involved to be properly accounted for as their requirements are recognized. All materials problems cannot be completely resolved in the design stage, but those that cannot should be reduced to secondary importance and greatly simplified by this prior attention. Thus production engineering and manufacturing can be entered into with a higher degree of assurance and engineers who are constantly working with materials will become "before the fact" diagnosticians, rather than "after the fact" trouble-shooters.

Design Co-ordination: This philosophy of materials selection and specification does not exclude the possibility or desirability of co-ordination of materials problems between the design agency and the production engineering agency. The relationship between the two must be very close. Neither the designer nor production engineer can be expected to cope with all of the complexities surrounding present-day materials, but much is accomplished when materials are selected and developed with the design. This is where the need for materials and the requirements for specifications have their origin. This is where fundamental control has its beginning. Countless specifications and controls have been written and many, because of the firm groundwork from which they originated, continue to serve, but these like so many of their obsolete predecessors are subject to revision and change if new design presents new specifications requirements.

Before any material can be selected or specified, consideration must be given to certain criteria which govern a choice. The degree of consideration will vary, but consciously or unconsciously, the following factors must be invariably weighed in the mind of the engineer if he expects to make a

wise decision on material selection.

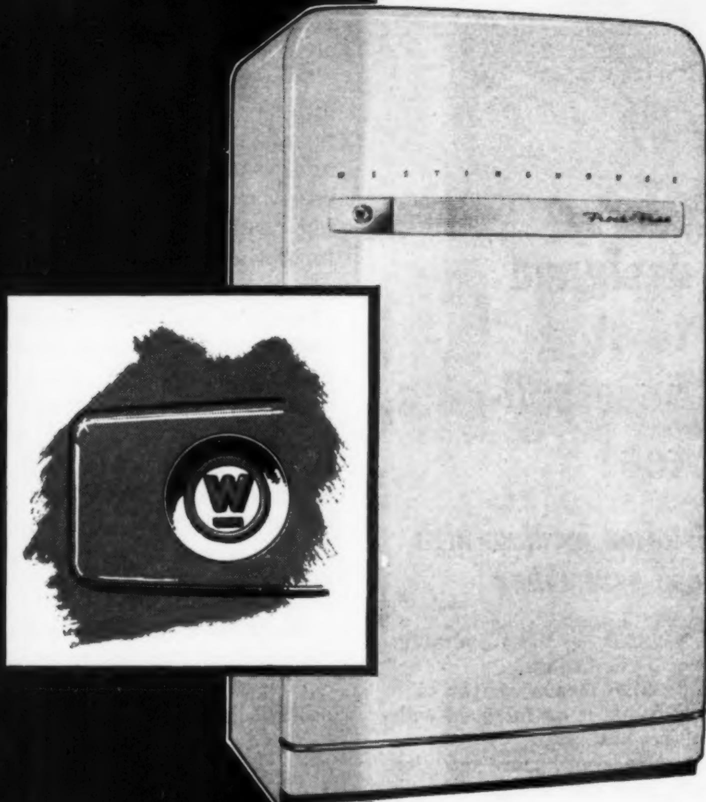
1. Engineering requirements
2. Manufacturing requirements
3. Availability
4. Economics

There are many more factors that could be enumerated, but in the final analysis these four items furnish a basic pattern for thought.

Engineering Requirements: The designer's first concern is usually with physical properties which can predetermine material selection on the basis of strength and functional characteristics and also constitute the framework for material specifications. His next thoughts are probably of service and environmental conditions with their related hazards, and finally he may try to evaluate expected performance. Then he decides on the materials to use. Several materials are selected for one job and these are narrowed down to an ultimate choice as the design progresses. Only one material may be chosen for a job and compensation for some of its inadequacies may be made by changing the design if warranted. If the designer's approach to the problem is sound, his selections will be essentially correct.

The era of aeronautics permitted the relationship between strength and weight to reach full significance. There is a world-wide trend toward the use of lightweight, high-strength materials in almost every type of industry. Supersonic flight continues to impose new and more demanding requirements on materials; ultramodern electronic equipment provides additional demands, and even a simple tool like the kitchen knife calls for more than just a piece of steel for a blade and a piece of wood for a handle. These constant changes are making the materials situation more complex and are resulting in the discoveries of new materials that are capable of fulfilling not one but several engineering requirements. One important fact is that these materials must be controlled more closely by specifications than ever before — they have to be to prevent normal complexity from becoming a night-

from advance-design *Solenoids* by Soreng



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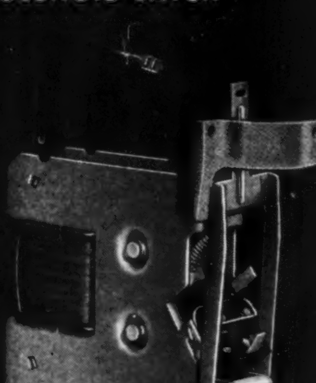

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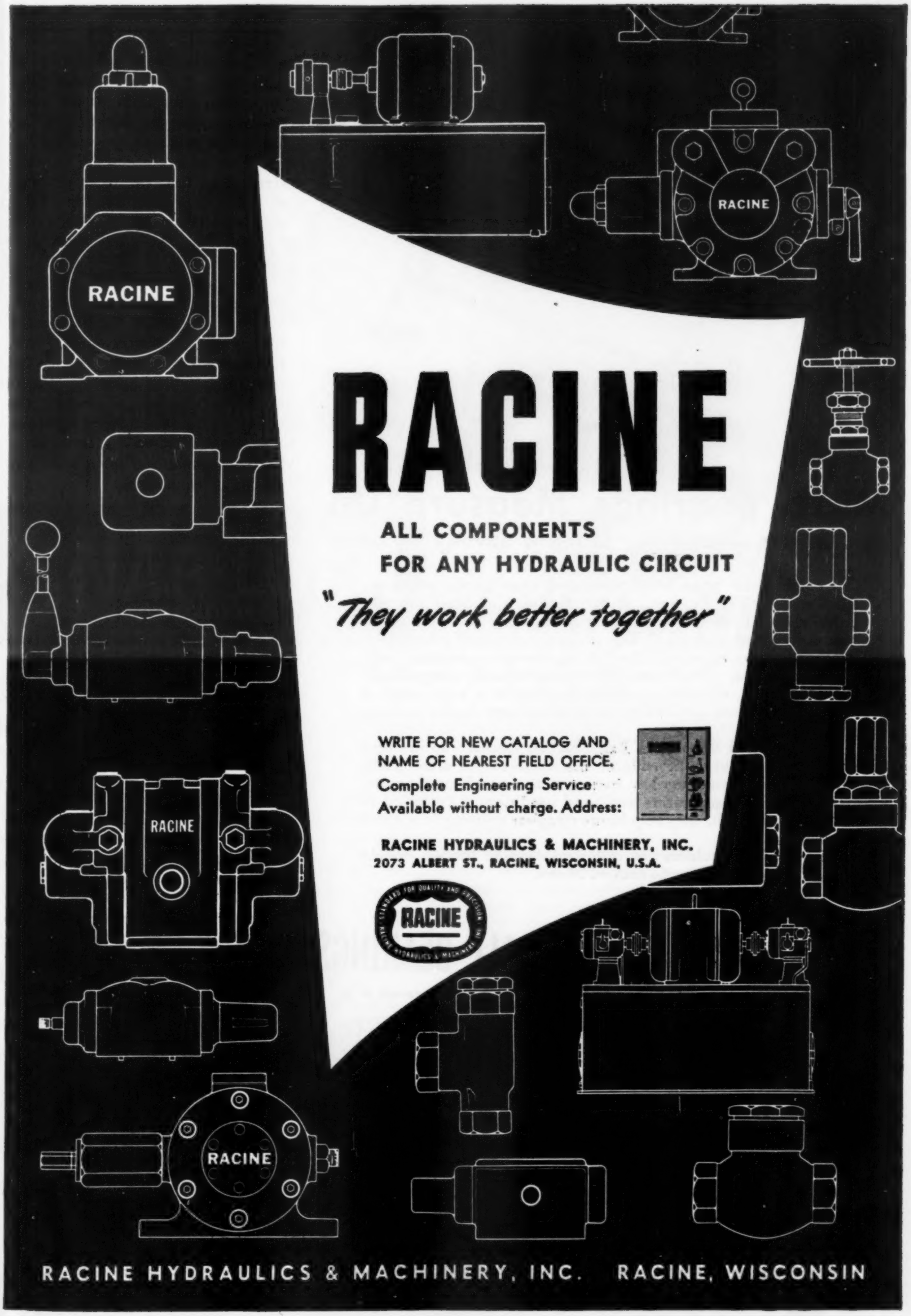
mare. Engineering properties fundamentally dictate the choice of materials, and the design engineer must recognize that care in selection and specification of materials is as important as the dimensional preciseness of his design.

Service and environmental conditions have their influence on materials selection and specification, and usually they must be considered simultaneously with engineering properties. When several materials are available for the same job, these two factors contribute greatly to making a choice. Here, compatability of one material with another must be analyzed; strength versus elevated temperature must be considered; weathering and corrosion resistance must be given attention. These are but a few of the problems that can be presented by service and environmental conditions.

Manufacturing Requirements: Satisfied is the engineer when he can say, "Here is a material that fulfills all of my immediate requirements for engineering properties, service and environmental conditions, and performance." However, the wise engineer fully realizes that he has only completed one hurdle. Eventually he has to decide whether the components of his design are to be machined, cast, forged, extruded, swaged, spun, welded, brazed, soldered or mechanically joined. He has to consider how manufacturing will reach to his choice of materials. Will they lend themselves to the manufacturing techniques with which they will be associated, such as machining, fabricating, processing and assembly? Obviously, no engineer wants to throw an unmachinable material to the shop for precision machining, nor does he want to give them a material to weld if it cannot possibly be welded.

In order to do a more thorough job of selecting materials, the engineer must weigh his selections based on his own immediate requirements against their suitability for manufacturing operations. If a material can fulfill the requirements of both engineering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and manufacturing, completely or sufficiently for the intended purpose, then there can be no question regarding its desirability for use. Specifications, governing the material selected, should be constructed so that the basic criteria, as well as pertinent factors affecting manufacturing, be embraced but not to the extent that they cannot be met.

More satisfied is the engineer who knows that his material selections will meet manufacturing requirements as well as his own. His design has a better chance for survival. But he also knows that all his efforts are wasted if the selected materials are unobtainable or restricted to the extent that he cannot use them with any degree of confidence.

Availability: Materials on the critical shortage list can only be used sparingly and sometimes not at all if their job has no priority —this must be considered. Restrictions on imports of basic raw materials affect availability. Material may be available but not in the desired form, size or condition. A decision has to be made whether to choose it anyhow and rely on the shop to do extra work to make it usable or choose another material which approximates the original choice but is slightly less desirable. The material may not be available at all. These factors must all be ironed out before the material is specified on a drawing, or very shortly a flood of engineering change orders and material deviations confront engineering. Availability can very often make the selection of materials more difficult than the engineering and manufacturing requirements.

Specifications themselves can impose restrictions on availability through misuse, distorted requirements, and confusion resulting from duplication. These contribute to the problem of finding specification material on the open market. The demand is greater than the supply for many items. This condition has developed with the increased application of government specifications and industry's growing appreciation of specification

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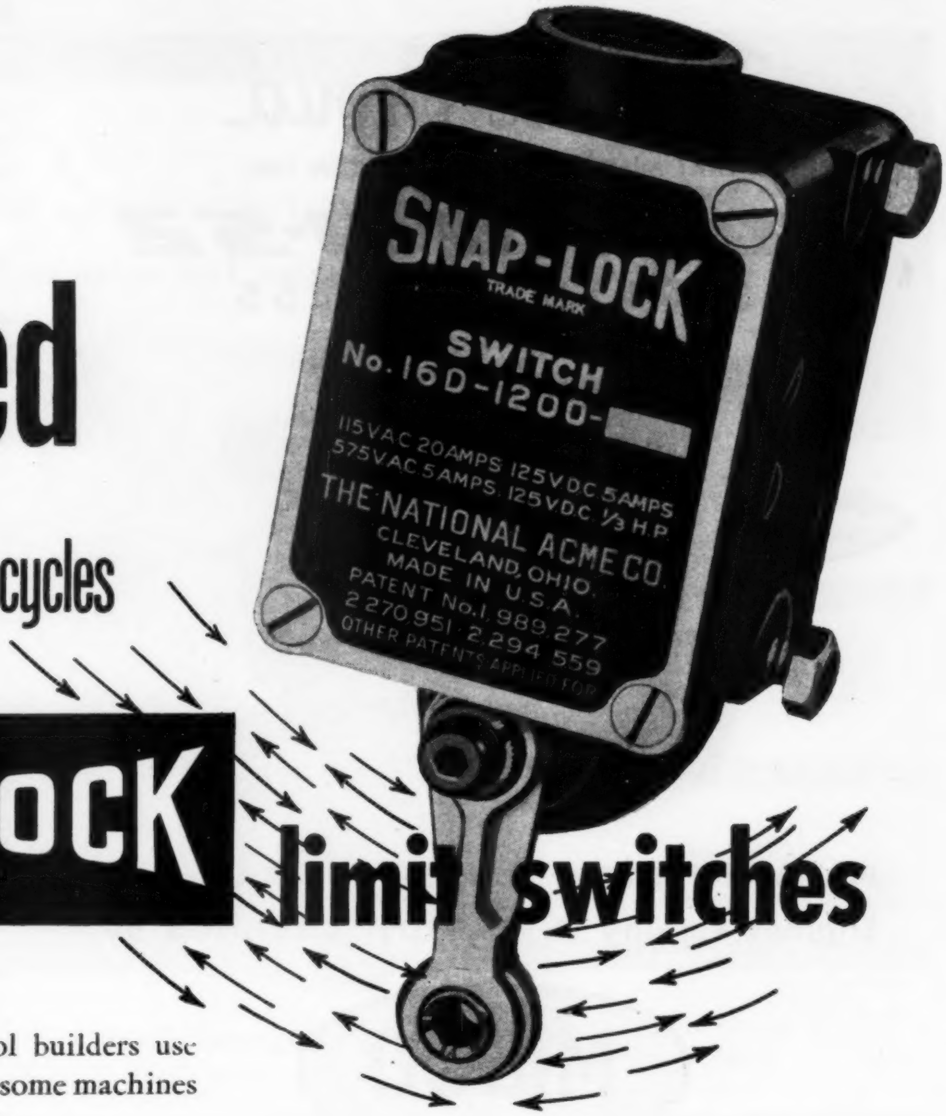

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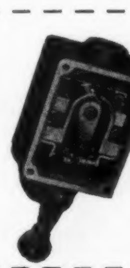

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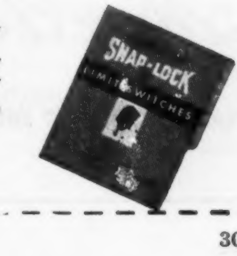

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materials. Vendors of material are affected by specifications and this effect is likewise felt by the consumer.

Economic Considerations: Thus, the very instrument that provides control can complicate the selection of materials. A material specification must fulfill its obligation to product design, but it should not be such that it will unreasonably hinder or restrict procurement. The economy exercised in the selection and specification of materials probably has more profound effect on the financial structure of a company than most people realize. To detail the economic aspects of material selection and their effects in an overall program would almost be a hopeless task, but one thing is certain—economy, whether it may be tangible or intangible, must be practiced and false economy must be guarded against when the best material is to be selected for its intended purpose. The judicious formulation and application of specifications will do much toward accomplishing this; they are the life insurance of a product.

From an address entitled "How to Select and Specify Materials" presented at the First Basic Materials Conference held concurrently with the First Exposition of Basic Materials for Industry in New York, N. Y., June 1953.

Management's Responsibility To Engineering

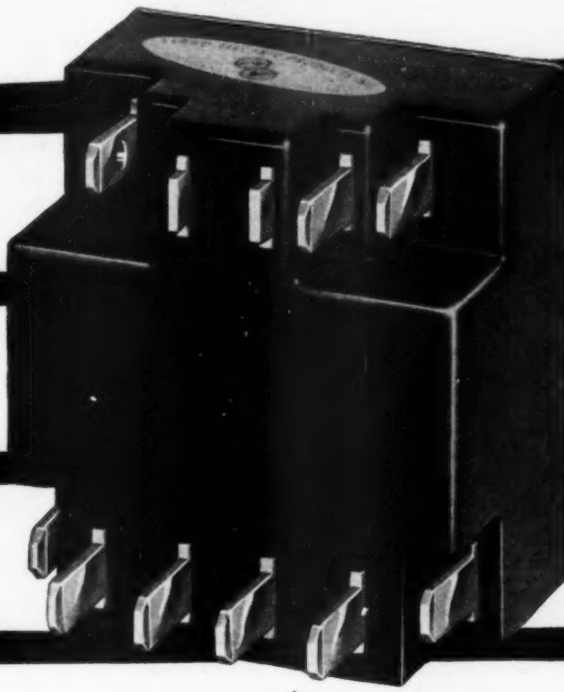
By W. R. Woolrich

Dean of Engineering
University of Texas
Austin, Texas

THROUGHOUT the past 183 years of professional engineering endeavor, during which there has been general concerted effort of engineers to achieve professional distinction, we also find some minor attempts to impede professional growth. No force from within, however, has formerly posed as great a threat as does the present activity of the Engineers and Sci-

LOOKING FOR Something New IN SWITCHES?

(This one helps SELL room air conditioners)



POSITION I
"VENTILATE"
(Evaporator, on full)

POSITION II
"EXHAUST"
(Condenser, on full)

POSITION III
"COOL and EXHAUST"
(Evaporator, on low; Condenser, on full; Compressor, on)

POSITION IV
"DEHUMIDIFY"
(Same as position III)

POSITION V
"SUMMER COOLING"
(Evaporator and Condenser, on full; Compressor, on)

POSITION VI
"SLUMBER COOLING"
(Evaporator and Condenser, on low; Compressor, on)

POSITION VII
"HEATING"
(Evaporator, on full; 1250-watt heater, on)

POSITION VIII
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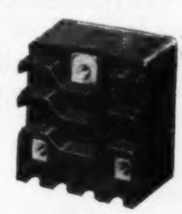
Special 8-POSITION switch
MODEL 190

The designing and building of *circuit selector switches* is an important part of Ferro's business. And we're not too big to fuss with *your* special requirements, provided the potential production volume justifies the effort—*both* for you and for us.

Whether your problem is one of space and compactness or the best handling of an intricate wiring pattern, you'll find interested, competent "first aid" here at Ferro. You'll find also that the finished product is *right* due to our small labor turnover and rigid testing and inspection system.

Illustrated above is one of our more elaborate switches. Shown at the right is a simple, 3-position model. Somewhere between the two may be the special switch you are looking for. *Why not write* and tell us your requirements?



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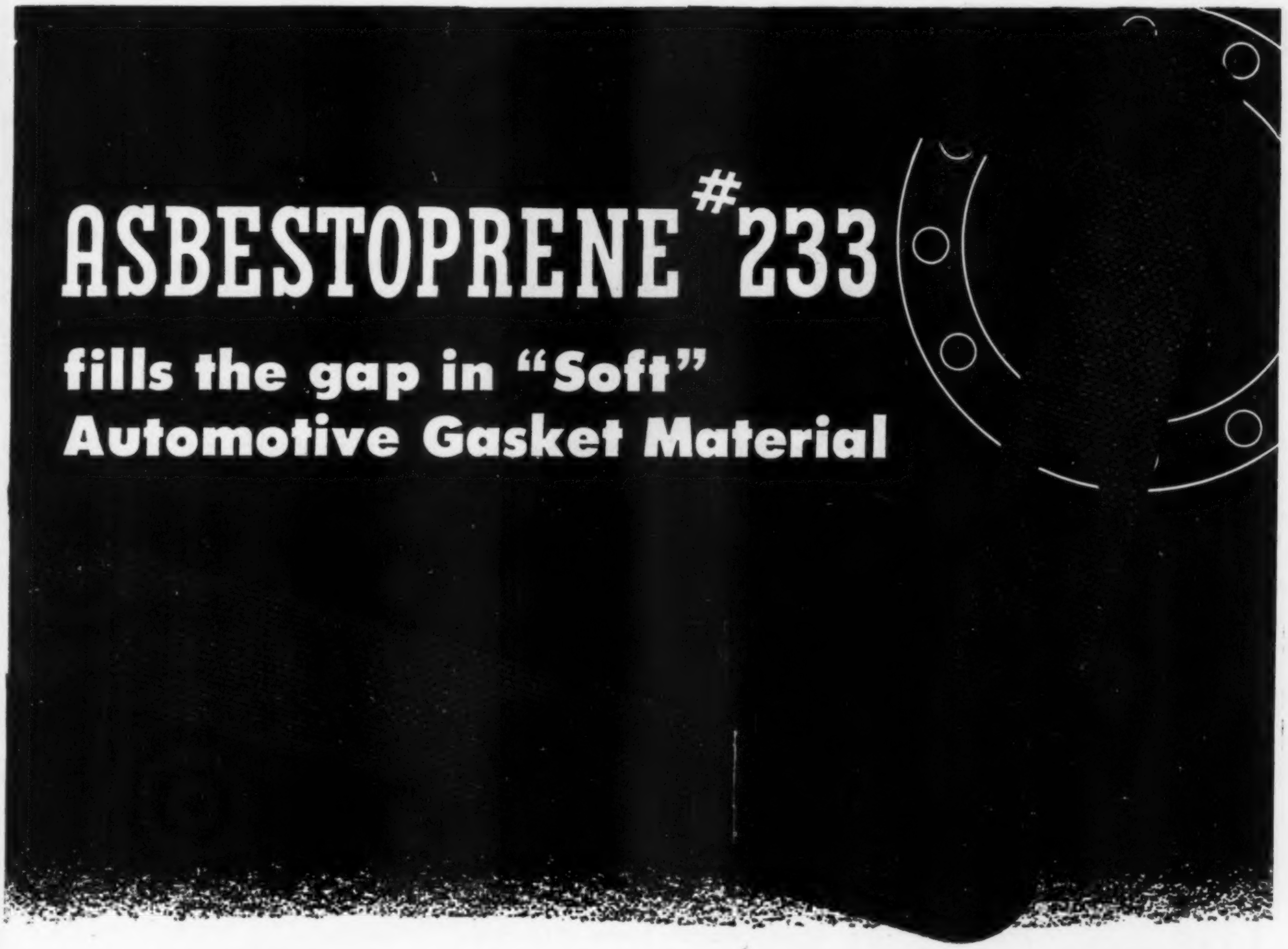
entists of America. They are now attempting to gain the ear of the U. S. Congress and the Secretary of Labor to the effect that their organization may be named "to speak for the approximately 400,000 employed nonsupervisory professional engineers and scientists who are so vital to our national defense and to our way of life."

Printed statements by President Joseph Amann of the Engineers and Scientists of America union in the supporting labor journals give definite evidence that the union seeks to place the average engineer in a class with tradesmen and definitely on the workman's side of the collective bargaining table.

I would like to re-emphasize the editorial statements given through the *Journal of Engineering Education* in the April 1953 issue: "The engineer of less than average ability can generally be given a salary lift by a collective bargaining agreement. In the levelling off process he gains at the expense of those who are above the average. When a union is first established there may be even a higher average wage than could have been obtained by individual contract by superior engineers. But the future is not too bright in collective bargaining for the *advancement of men of creative genius and individual professional abilities.* Collective bargaining is never inspirational to the man of creative talents, and recognition within a union cannot be readily afforded him other than as a part of a collective group."

Professional Recognition: On the other side of this question there is a large segment of American industry that has failed to be realistic in its adjustment of salaries and wages of its employed professional engineers. The salaries for beginning engineers have risen at a much faster ratio than have the increments of advancement for qualified engineers of five, ten, and twenty years of service in some companies within this unrealistic industry segment to which I refer. Actually, in some instances the starting salary curves for recent graduates have intersected the average salary curve of men of some years



New Asbestos-Neoprene Sheet Packing is Both Compressible and Heat Resistant

Where heretofore you've had to sacrifice compressibility or heat resistance in standard sealing materials for oil, greases, water, gasoline, etc., now ASBESTOPRENE #233 combines these properties to a very useful and satisfactory degree. In addition, ASBESTOPRENE #233 shows marked improvement in other characteristics over the usual compromise specifications such as gelatin-treated paper or cork. It is better in corrosion and fungus resistance . . . in torque loss . . . and dimensional stability.

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Your own tests will convince you ASBESTOPRENE #233 is the best packing in its class for recommended applications. Get all the facts through your Victor Field Engineer, or by direct inquiry to the Victor Company.

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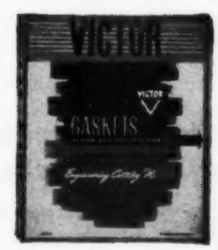

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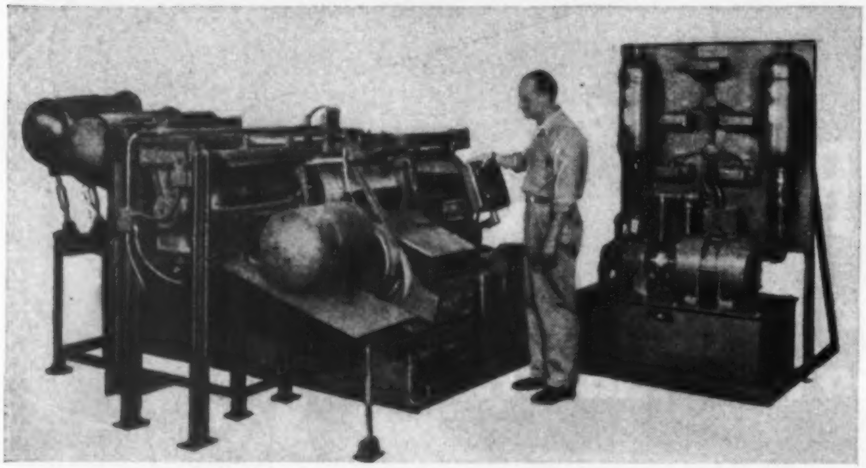
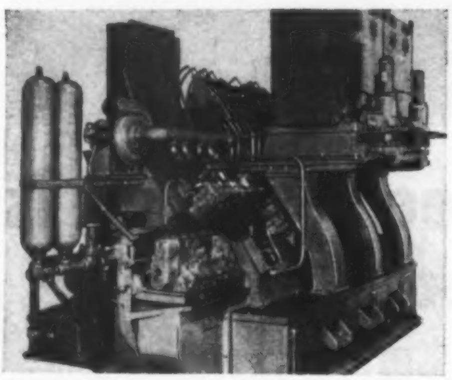




of service and experience within the engineering profession. It is the American industrialists of this negligent segment who furnish bonafide reasons for some union or other group to challenge the administrative competence of the managers concerned.

The employer of the professional engineer, whether he is with a University, a Federal or other government agency, or an industrial concern, should provide the incentive and recognition for advancement within the profession whenever possible. If industry and government are going to avert the growth of unionism within professional engineering circles, then the compensation offered for engineering service must be commensurate with the abilities and contributions of each individual within the profession. *It is essential that the salaries of men of accomplishment and years of service shall be an incentive to the men entering the profession.* This calls for a wider spread in compensation between the salaries of the beginner and the engineer of several years of competent service. This spread should be commensurate with their respective values to the employer.

We within the engineering profession, together with the employers of engineers, must face this problem with understanding and positive constructive action. Those within the union circles who would speak for us at the conference table are alert to the weaknesses of the salary scales of some industrial groups and are always ready to move in.

The most effective answer to unionism is a full recognition by the employer of each individual as a professional engineer. The professional engineer is usually a man who prefers freedom of thought and action to that of herd classification, but he should—and does—expect adequate consideration for his specialized services.

From an address entitled "Education for Professional Engineering Responsibilities" presented before the Annual Meeting of the American Society for Engineering Education at the University of Florida, June 1953.

New Machines

Communication

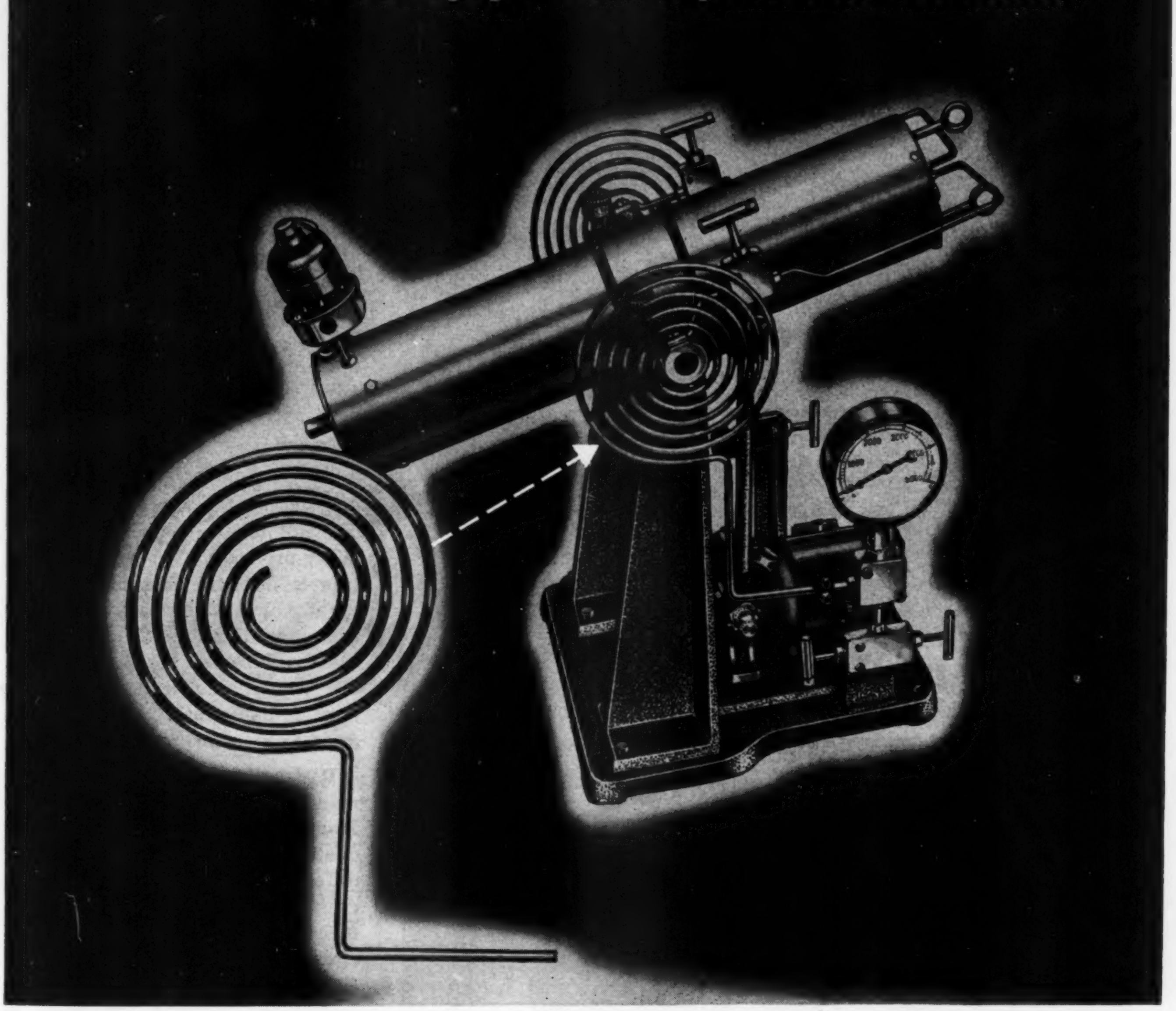
Tape Recorder: Portable Magnematic attains frequency response of 50 to 1500 cycles at tape speed of 7½ ips. Six models available in three speeds with playing time up to 1 hour. Features pushbutton control, solenoid-operated capstan drive to start and stop tape travel within 1/20-sec, relay-operated modified Geneva movement to control high-speed rewind and 60 ips fast-forward functions. Interlocking operation prevents tape break or spill. Uses 5-in. plastic or metal reels of standard ¼-in. tape. Operates on 110/125 v, 50/60 cycle ac; consumes 90 w. Size, 8½ by 11 by 9 in.; weight, 19 lb. *Magnematic Div., Amplifier Corp. of America, New York, N. Y.*

Television Receiver: Beverly consolette 21-in. screen model is made of natural-finish straight-grained birch; has a wrought iron and birch stand. Dimensions of the set are 25 in. wide, 22⅝ in. deep and 23⅜ in. high; 37 in. high with stand. *Motorola Inc., Chicago, Ill.*

Portable Radio: Automatic shutoff disconnects batteries when the radio is plugged into an ac or dc outlet. Begins to play as soon as it is turned on. Has improved superheterodyne circuit and precision tuning. Civilian defense 640 and 1240-kc frequencies are marked on dial which is located on the top of the case. Carrying handle folds to the side when not in use. Available in maroon, green or gray cabinet. *Admiral Corp., Chicago, Ill.*

Airborne Tape Recorder: Nadar unit utilizes miniature electronic parts to log events occurring during an airplane or missile flight. Can record information during a 10-hour flight and can then be run for another 10 hours nonstop, with previous information being erased as new information is recorded. Operates on 28-v dc. Uses ¾ in. wide plastic magnetic tape. Size, 8 by 9⅞ by 9⅞ in.; weight

Windpipe for a peaceful H-bomb



You can call it an H-bomb if you like. Some people do. Technically, it *is* a hydrogenation bomb or, simply, a superpressure vessel. Uses for it are legion but not warlike.

Researchers in rubber, plastics, pharmaceuticals, petroleum, and solvents pilot their hydrogenation, oxo-reaction, polymerization and other jaw-breaker problems in its gizzard—under high pressure.

To speed-up chemical reactions, the vessel is rocked 38 times a minute. Pressurizing a see-sawing chamber is in itself a neat trick: when pressures of the order of 60,000 p.s.i. are required you might become discouraged or concerned.

American Instrument Company, Inc., maker of Superpressure equipment, introduces the pressure through a spiral windpipe. They had some trouble through tube breakage but that was before they tried Superior 304 Cold-drawn Seamless Stainless Steel Tubing. Superior gives them just the flexibility and fatigue strength needed for cyclic operation, plus chemical resistance and burst pressure to spare. Spirals of Superior 304 are easier to bend, last longer. And American Instrument reports them dimensionally uniform, smooth and free from surface imperfections.

If you haven't found small tubing in the analysis, size, and finish you want, or if you think there's little difference in tubing or tubing experience, Superior invites your inquiry. Superior Tube Company, 2010 Germantown Avenue, Norristown, Pennsylvania.

Round and Shaped Tubing available in Carbon, Alloy, and Stainless Steels; Nickel and Nickel Alloys; Titanium, Zirconium, and Beryllium Copper

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All analyses .010" to ⅝" O.D.
Certain analyses (.035" max. wall) up to 1⅜" O.D.

New Machines

18 lb. *North American Aviation Inc., Los Angeles, Calif.*

Materials Handling

Fork Truck: For placing and removing dies from presses. Handles 1500-lb dies, 650-lb adapters, 400-lb trimmer dies and 50-lb die inserts. Capacity is 4000 lb. Has 42-in. forks and 42-in. boom which is used when dies cannot be handled easily with the die-pulling device. Twin drum die-puller is operated by a separate electric motor. Truck is 83 in. high and has a 66-in. nontelescoping lift. Boom and forks are removable, and the truck can be used as a standard fork truck. *Elwell-Parker Electric Co., Cleveland, O.*

Slat Conveyor: Pusher type handles hot billets and similar heavy, bulky products. Conveyed material travels on bars that are clamped to a pair of power-driven chains travelling in the side rails of the conveyor frame. Available in various widths and frame pitches to meet individual requirements. *Sage Equipment Co., Buffalo, N. Y.*

Hydraulic Loader: Shoveloader mounts on Case, Minneapolis-Moline, Shepherd and Oliver industrial tractors. Available in six different models and with eight accessory attachments. Basic attachment is a material bucket for materials loading and unloading. Loader is equipped with double-acting hydraulic cylinders for precision control of loads in any direction. Reaches 66 to 72 in. in front of radiator and digs below tractor level. All actuating parts are located in front of the cab. *Baker-Lull Corp., Minneapolis, Minn.*

Hand Truck: Fabricated of magnesium, the Magcoa weighs 18 lb and has a capacity of 500 lb. Features all-bolted construction. For use on delivery trucks and for general factory, warehouse and loading dock use. *Magnesium Co. of America, East Chicago, Ind.*

Air Hoist: Roller-chain type, powered by rotary-vane air motor. Lifts 1000-lb loads at a rate of 40 fpm. Rate of lift is variable from 0 to 40 fpm, and a 1600-lb load can be lifted. Loads can be

New Machines

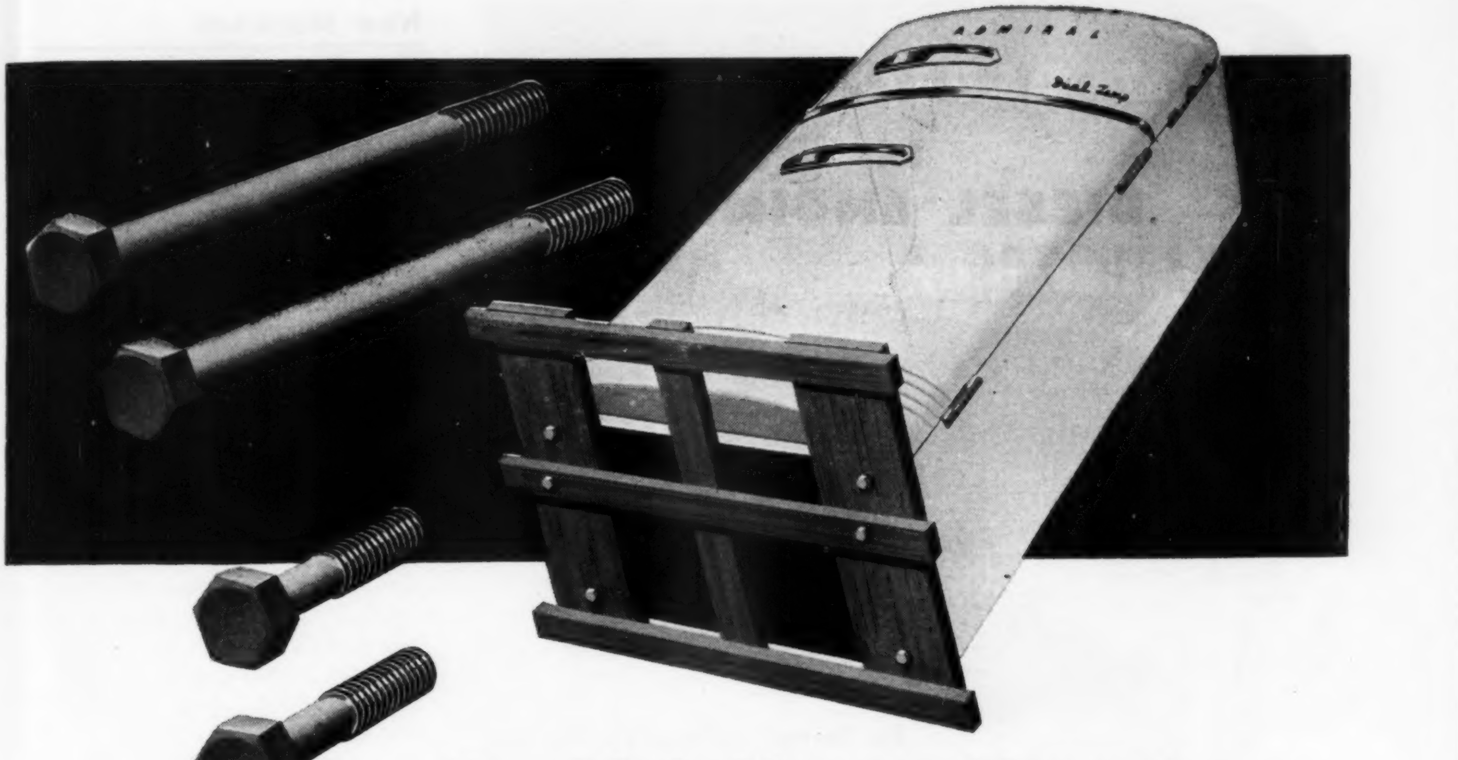
lowered at speeds exceeding 100 fpm. Safety brake automatically locks when control is released, so that load will remain suspended if the hose is accidentally disconnected or the air shut off; however, load can be released by throttle control and lowered. Other specifications include: standard lift, 8 ft; recommended hose size, $\frac{1}{2}$-in.; weight of unit, $27\frac{1}{2}$ lb; minimum hook to hook distance, $13\frac{1}{4}$ in.; overall length, $10\frac{1}{8}$ in.; diameter of body, 5 in. *Aro Equipment Corp., Bryan, O.*

Pallet Tiering Truck: Hydraulically operated Pallet Stacker has $\frac{3}{4}$-hp motor-driven pump; will tier pallet loads as high as 115 in. Rate of elevation is 8 fpm. Base forks have six separate adjustments for handling pallets from 30 to 48 in. wide. Mast extends $138\frac{3}{4}$ in. when elevated, 83 in. collapsed. Rated capacity is 2000 lb; truck weighs approximately 1300 lb. *Raymond Corp., Greene, N. Y.*

Carton Closer: Model 518, designed for use with the manufacturer's Convey-O-Mat carton set-up machine, can be used to close cartons on any production line. Distance between set-up and closing units can be varied to meet cartoning requirements. Portable; occupies 6 sq ft of floor area. *Machinery Mfg. Co. Inc., Los Angeles, Calif.*

Metalworking

Punch Shaper: Essex 32A unit machines a blanking punch, hob or profiled part directly from the solid in one operation. Special punch holders or plates are not needed. Can produce fine surface micro finish, controlled through microscopic readings. Stroke of the cutting tool is like that of a conventional shaper set vertically and is adjustable from $1\frac{1}{2}$ to $4\frac{1}{2}$ in. However, upon completion of the predetermined straight stroke, the rocker arm operates in a swivel motion, finishing the desired radius. Punch can be made a few thousandths oversize, then sheared into the die to a depth of approximately 1/32-in. Then, using the microscope, it is finished to the exact



How Admiral freezes rising costs—saves $15,630 with Townsend Fasteners

Efficient planning all along Admiral production lines right down to the shipping platform brings you excellent values in home appliances and other Admiral products at the lowest possible costs. No detail is too small—no part too insignificant to come under the scrutiny of their engineers who constantly seek ways to improve their products.

Take the big Admiral two-door Dual-Temp home freezer-refrigerator, for example. The six bolts that hold it to the shipping crate could seem to be unimportant. Yet, a Townsend engineer found that by the Townsend method of producing these fasteners, Admiral could save $13.37 per thousand on one and $6.34 per thousand on the other bolt. The total annual cost reduction of about $15,630 presents a definite opportunity to put greater value in the refrigerator without increasing the price to the consumer. Such savings are not unusual with Townsend's fasteners and parts.

In today's competitive markets, many industries rely upon Townsend as "The Fastening Authority" to help improve products, reduce unit and assembly costs. Every day Townsend engineers work with manufacturers of appliances, radio and television receivers, automobiles, aircraft, electrical equipment, farm implements, railroad rolling stock—even toys. In fact, Townsend serves every industry that needs to fasten materials together quickly — economically — permanently.

With a background of 137 years of cold-forming experience—with the capacity to produce sixty-million parts every day—with more than ten thousand standard and special items to choose from—Townsend engineers give you the most unbiased recommendations. Ask to have one call and discuss your fastening problems—or, send us samples and sketches of parts you want improved at substantial savings. We will give you estimates without obligation.

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Sales Offices in Principal Cities

Cherry Rivet Division • Santa Ana, California

THE FASTENING AUTHORITY—Experience: *over 137 years*—**Capacity:** *sixty-million parts daily*—**Products:** *over ten-thousand types of solid rivets—cold-headed parts—Cherry Blind Rivets—Twinfast Screws—self-tapping screws—tubular rivets—locknuts—special nails—formed wire parts.*
Plants: *New Brighton, Pennsylvania—Chicago, Illinois—Plymouth, Michigan—Santa Ana, California.*

New Machines

contour of the sheared in section to an accuracy of ± 0.0005-in. Special table-shaping attachment permits swiveling the dividing head from 0 to 15 degrees toward the tool and 5 degrees to right and left so that the operator can put relief on punches or draft on hops. *International Machinery Div., British Industries Corp., New York, N. Y.*

Adjustable - Speed Drive: For Rogers "Perfect 36" vertical turret mill. Provides table chuck speeds up to 212 rpm. Independent speed control box which may be located for operator convenience is equipped with rheostat control; has start, stop and jog pushbuttons; sets and changes motor speeds and provides a ratio of more than 16 to 1. Speeds may be changed while machine is in operation. May be modified to meet JIC specifications. Mill has adjustable five-position main vertical turret for tool settings including boring, drilling, reaming, turning and thread cutting. Square turret indexes to eight positions. *Rogers Machine Works, Buffalo, N. Y.*

Acetylene Torch: Performs light heating, brazing or soldering operations. Operates on low pressure. Uses acetylene gas from a Presto B tank; develops a temperature of 2800 F. Can be used for all kinds of seam soldering. Soldering copper reaches proper heat quickly and holds that heat until turned off. Two long tips, one straight and one curved tip are supplied. *Torit Mfg. Co., St. Paul, Minn.*

Grinding Attachment: Simultaneously revolves and moves work longitudinally back and forth. Standard spacer gears with a follower pin for each flute to be ground provide indexing for 2, 3, 4, 6, 8, or 12 flutes. Special spacer gears available. Number of back-and-forth movements per revolution of work is controlled by number of follower pins used in spacer gear. Permits grinding of form and/or radial relief, tapered cylindrical and straight cylindrical. Cutting tool to be produced or reworked is held in collet or between dead centers and revolves on its own axial center. Where full length of spiral cutting tools is to be

ground for both form and radial relief, attachment travel is similar to an OD grinder. *Detroit Reamer & Tool Co., Detroit, Mich.*

Power Bender: Redesigned Di-Acro unit for forming simple and complex bends in round stock, tubing, angles, channels, molding, strip stock, extrusions and other material forms. Makes either clockwise or counterclockwise bends. Table is made of ribbed alloy casting which provides high strength during bending and allows integration of gear housing into the casting, assuring positive alignment at all times. Foot controls are provided in addition to hand controls. Cabinet is made of ¼ to ½-in. steel plate. Motor is 3-hp. *O'Neil-Irwin Mfg. Co., Lake City, Minn.*

Surface Grinder: Capacity: transverse, 6 in.; longitudinal, 12 in.; vertical, 13½ in. under 7-in. diameter wheel. Takes standard 7 by ½ by 1¼-in. wheel. Has antifriction ball and roller-bearing motorized spindle. Spindle speed, 3500 rpm. Vertical and transverse wheel graduations are in 0.001-in. Has V and flat ways. Floor space required, 40 by 45 in.; weight, 750 lb. *Bridgeport Grinding Machine Co., Bridgeport, Conn.*

Cutoff Machine: Model M75 can be used as a regular cutoff machine or converted to such use as freehand cutting of gates and risers from nonferrous foundry castings. When used as a cutoff machine either abrasive blades for cutting ferrous materials or semi-high-speed steel saw blades for cutting nonferrous materials may be used. Capacity as a cutoff machine, 2½-in. solids and 4-in. pipe, tubing and structurals. Equipped with 7½-hp motor. *Stone Machinery Co. Inc., Manlius, N. Y.*

Threader: "Precision-Pak" unit designed for high production and precision. Features direct precision lead, rapid tool spindle return, variable speeds and low maintenance cost. Spindle is driven by 3-hp motor. Quick-change lead screw and collapsible lead nut head mounted directly on end of spindle maintain precision lead throughout 4-in. feed travel. Lead nuts are automatically disengaged at end of feed travel and spindle return is

accomplished by a planetary gear clutch. Unit can be mounted in many positions, such as horizontal, vertical or angular in single, double end or multiples. *Sheffield Corp., Dayton, O.*

Centerless Grinder: Diversimatic grinds bearings, bushings, cap screws, shafts and formed parts of two or more diameters or contours. Features removable grinding wheel, spindle quill; combined double-row superprecision ball and roller bearings permanently sealed and lubricated. Wheel clearance, 0 to 1½ in.; speed of grinding wheel, 1600 rpm. Regulating wheel is infinitely variable from 30 to 300 rpm. Crush Forming attachment permits precision grinding of small formed parts that otherwise could not be ground between centers because of work deflection. Floor space required, 48 by 36 in. *Van Norman Co., Springfield, Mass.*

Broaching Machines: Ten-ton, 60-in. machine broaches three scallops with each pass of the ram; 10-ton, 90-in. model broaches a single dovetail slot with each stroke. Automatic—the work shuttles into the cut and the ram starts. At the completion of the stroke, the fixture recedes from the broach cut and the ram returns to starting position. Part is then automatically indexed one increment and the process is repeated until all slots or scallops are completed. Fixture then recedes to loading position. Pneumatic or mechanical ejection for finished parts is optional. *Colonial Broach Co., Detroit, Mich.*

Office Equipment

Rotary File: Motorized card file is available in two sizes which accommodate approximately 62,000 or 85,500 cards of 3 by 5-in. size. Other models can take 4 by 6-in. or 5 by 8-in. cards. Average time required to bring desired card directly in front of operator, three seconds; complete revolution takes nine seconds. Cards are set in individual trays which are removable, with a follower block in each. Card trays 8 in. long carry approximately 800 cards each. Cards may be read without being re-

(*Continued on Page* 321)

New Machines

(*Continued from Page* 318)

moved from proper sequence. From 4000 to 17,500 cards are always open before operator and within reach from front to back and left to right. Operation is controlled by either pushbuttons on the right-hand side of file or a special foot pedal switch. Size: Model 45, 35½ in. deep, 54 in. wide, 37½ in. high; Model 60, 35½ in. deep, 71 in. wide, 37½ in. high. *Ferris Business Equipment Inc., Stratford, Conn.*

Adding Machine: Adds, substracts, multiplies and divides. Features operating speed of 190 cycles per minute, "touch-contoured" keyboard with alternating rows of flat and concave key tops to facilitate touch operation and reduce errors, two-tone gray and green finish to reduce eyestrain and sound-proof construction for quiet operation. *Clary Multiplier Corp., San Gabriel, Calif.*

Packaging

Case Sealer: Packomatic Model D applies glue in various spot patterns and seals all types of paper cases. Glue is applied to the inner surface of outer case flaps in spots at ¾-in. centers. Brass glue applicators will not rust or corrode and can be cleaned or replaced quickly. Stainless steel glue supply tank with stainless screen filter supplies adhesive to applicators through transparent plastic tubing. Accommodates slotted containers 8 to 24 in. long, 6 to 16 in. wide and 5 to 18 in. high. Seals eight 12-in. long or four 24-in. long cases per minute. Powered by ⅓-hp motor. Occupies approximately 30 sq ft of floor space; weighs 1150 lb. *J. L. Ferguson Co., Joliet, Ill.*

Tape Dispenser: Automatic, pressure-sensitive Grip-A-Tab can be connected to any air line. Foot lever controls feed of predetermined lengths of tape. Model AF-92 dispenses cellophane, acetate fiber, plastic, paper and electrical grade tapes from ¼ to 1 in. wide, accommodates up to 5½ in. OD roll or multiple rolls; delivers any length between ½ and 5¾ in. Model AF-85 dispenses the same materials

plus filament, cloth and heavy paper from 1/4 to 2 in. wide in rolls, up to 6 1/2 in. OD, or multiple rolls, and delivers tape in any length between 1 1/2 to 9 in. Multiple strokes provide longer lengths. *Derby Sealers Inc., Derby, Conn.*

Plant Equipment

Filter Pumps: Model LSIN-5 is rated at 50 gph and Model LSIN-10, at 100 gph. Filter most acid or alkaline solutions from pH 0 to pH 14; may also be used for filtering nonaqueous solutions. Remove particles from 150 to 1 micron in size. Self-priming pump has a suction lift of approximately 15 ft, delivers 6 gpm on open pumping, develops 40 psi pressure. Filter tubes are available in cotton, dynel, porous stone or porous carbon. Motor is 1/4 hp, 110 v, 60 cycle, single phase. Size of both models, 14 by 16 by 12 in. high. *Sethco Mfg. Co., Brooklyn, N. Y.*

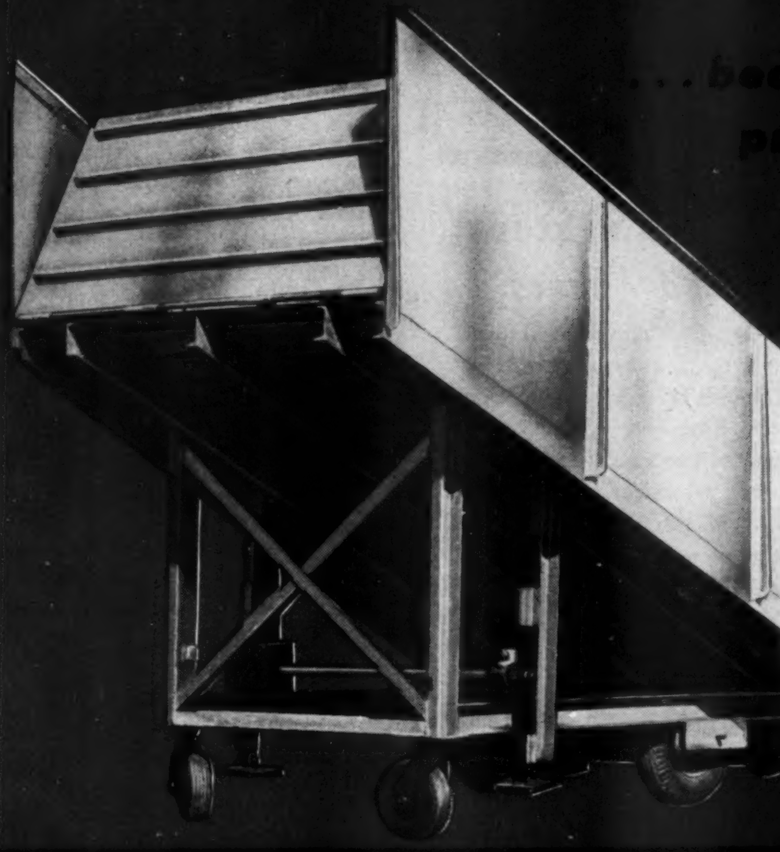
Battery Charger: Four-circuit Hobart Monitor for six-cell lead-acid or ten-cell nickel-iron motorized hand truck batteries. Four charge controls, the motor starting switch and the motor and generator controls are mounted in the monitor-top control cabinet as an integral part of the motor generator set. *Motor Generator Corp., Troy, O.*

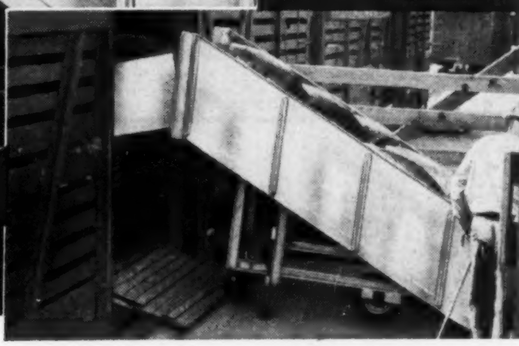
Pump: Type "DF" end-suction, centrifugal unit with capacity up to 1000 gpm and heads to 400 ft. Features doube-row thrust bearings, deep stuffing boxes, renewable case wearing rings and shaft sleeve with rubber sealing ring. Pumps water, condensate, solvents, brines, caustics, acids, organic liquids, hydrocarbons, gasoline, oils, etc. *Warren Steam Pump Co. Inc., Warren, Mass.*

Portable Tools

Metal Cutter: Redesigned "Little Wonder" nibbler incorporates a punch and die arrangement for cutting metals, including stainless steel, up to 14 gage. Cuts holes in tubes and ducts without damaging original contours; cutting pressure is applied only to the metal being

WHY ALUMINUM?





CASE NO. 37 PIONEER PORTABLE RAMP

By switching from wood to aluminum, the Pioneer Iron Works of Sioux City, Iowa, dramatically reduced the weight of these pig loading ramps from *half a ton* to only 200 pounds—a cut of 80%! Since these are portable ramps, moved from stock car to stock car when loading pigs, the importance of this weight reduction can be immediately seen. One or two men can now easily handle the ramps where formerly a crew was needed. Thus cost of operation is immediately lowered and the operation is speeded up.

Furthermore, aluminum does not rust, rot, mildew or otherwise deteriorate from exposure to weather. These aluminum ramps do not require painting or other protective maintenance—they are much easier to clean and thus more important savings are effected. These are the kinds of dividends you can realize by using aluminum.

In almost every industry a change to aluminum has provided manufacturing economies, improved designs and, at the same time, increased sales appeal. Ask Reynolds Aluminum Specialists to help you apply aluminum's advantages to your products and production.

Call the nearby Reynolds office listed under "Aluminum" in your classified telephone directory, or write to Reynolds Metals Company, 2521 South Third Street, Louisville 1, Kentucky. Also, write for complete index of Aluminum Design and Fabrication technical literature.

"Mister Peepers" returns September 13th on NBC-TV.

REYNOLDS ALUMINUM

MODERN DESIGN HAS ALUMINUM IN MIND

New Machines

cut. Fan cools motor when in continuous use. Available with a 220-v universal motor or regular 110-v motor. *Fenway Machine Co. Inc., Philadelphia, Pa.*

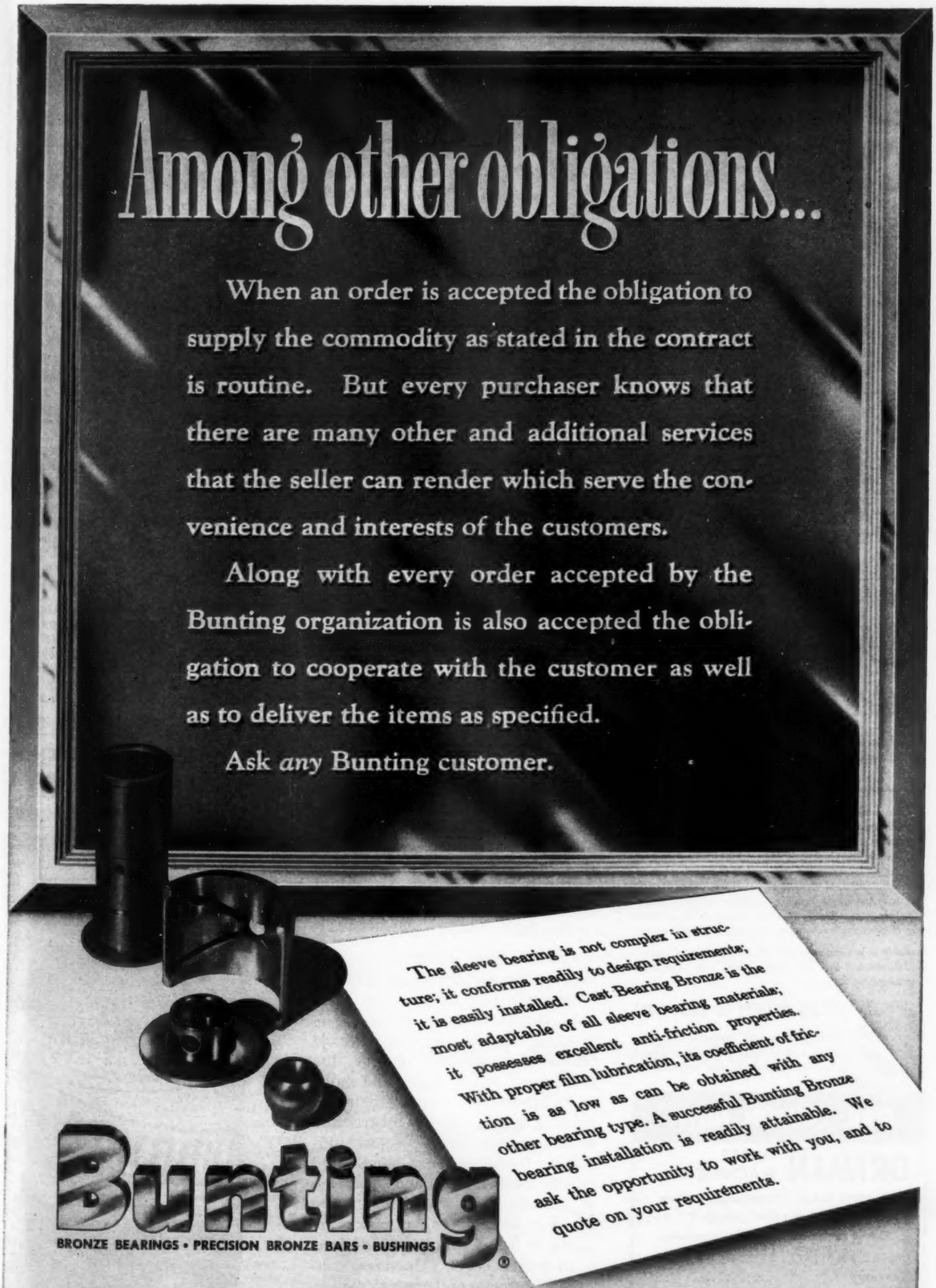
Electric Drill: Model 14-AU, 1/4-in. capacity, is built for continuous duty. Features ball-bearing construction throughout, aluminum alloy frame, Jacobs three-jaw geared chuck and universal motor. No-load speed is 2500 rpm. Operates on 115-v, 25 to 60-cycle ac or dc. Weighs 5 lb, is 10 1/2 in. long. *U. S. Electrical Tool Div., Emerson Electric Mfg. Co., St. Louis, Mo.*

Pencil Type Grinder: Air driven, will operate on pressures of 50 lb or more. Develops speed up to 110,000 rpm and therefore is adapted to Carboloy burrs. Exhaust air expelled from front end of the grinder blows dust away from the work. Used in die making, wood carving, fine etching, polishing and finishing operations. *Nu-Jett Products, Grand Rapids, Mich.*

Testing and Inspection

Gear Lead Checker: Sine-Line checks both lead and axial pitch on helical or herringbone gears from 18 to 48 in. OD with a maximum face width of 25 in. and a maximum shaft length of 96 in. Also checks parallelism of teeth on large spur gears. Either right or left-hand leads may be checked. Uses sine-bar principle to check leads from 0 to 80 in. Operated by either manual or power drive. Gears of varying face widths or diameters; but with same lead, may be checked without changing setting of sine bar. Consists primarily of a transverse table carrying the sine bar and a longitudinal table which carries the indicator. *Michigan Tool Co., Detroit, Mich.*

Comparator: Universal Electrolimit circuit and gage block comparator stand combined with the electronic recorder provides an instrument with 110,000 to 1 magnification. Range of the 11-in. scale of the recorder is 0.0001-in., with each primary division equal to 5-millionths in. and secondary divisions equal to 1-millionth in. Recorder has pen lifter and chart-drive


Among other obligations...
When an order is accepted the obligation to supply the commodity as stated in the contract is routine. But every purchaser knows that there are many other and additional services that the seller can render which serve the convenience and interests of the customers.
Along with every order accepted by the Bunting organization is also accepted the obligation to cooperate with the customer as well as to deliver the items as specified.
Ask *any* Bunting customer.
The sleeve bearing is not complex in structure; it conforms readily to design requirements; it is easily installed. Cast Bearing Bronze is the most adaptable of all sleeve bearing materials; it possesses excellent anti-friction properties. With proper film lubrication, its coefficient of friction is as low as can be obtained with any other bearing type. A successful Bunting Bronze bearing installation is readily attainable. We ask the opportunity to work with you, and to quote on your requirements.
Bunting
BRONZE BEARINGS • PRECISION BRONZE BARS • BUSHINGS
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switch so that under normal operation it is used as an indicating instrument. Measurements can be recorded on the chart paper, however, providing a permanent record. *Pratt & Whitney, Div. Niles-Bement-Pond Co., West Hartford, Conn.*

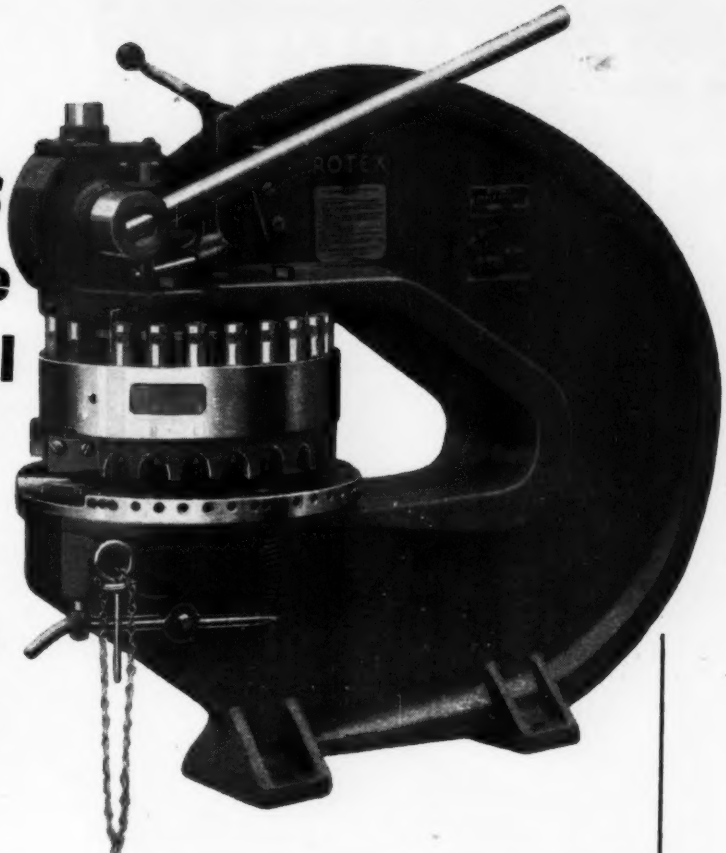
Woodworking

Circular Saw: Portable S-12 Air Saw operates on 90 psi air pressure. Cuts to a maximum depth of $4\frac{3}{8}$ in. Spur gear design of blade drive reduces arbor length between motor and blade. Throttle lever is located on the inside of the grip handle, and the saw blade is enclosed by a housing on top and a telescoping guard underneath. Hand screw and thumb screw adjust depth and angle of cut. Maximum thickness of material cut when the saw is set for a 45-degree bevel cut is $2\frac{7}{8}$-in. *Ingersoll-Rand Co., New York, N. Y.*

Radial Arm Machine: Model GA, replacing previous model GR, has a longer arm. Roller head carriage which supports the overhead cutting member rides on eight bearings. Four of the grease-packed, double-row ball bearings are set at a 90-degree vertical plane to absorb the extra upward thrust from rafter notching, compound angle cutting, etc., and the other four are set at a 45-degree plane to absorb the side thrust created through normal sawing action. *DeWalt Inc., Lancaster, Pa.*

Electric Saw: Heavy-duty, $8\frac{1}{4}$-in. capacity portable Model 108 can cross-cut a 1 by 10-in. plank in ½ sec. Safety features include telescoping guard, auxiliary guard which automatically covers the front teeth at any depth and built-in kickproof clutch that eliminates bucking if blade hits an obstruction or binds in the wood. Die-cast aluminum frame has loop handle and trigger switch; one-hand operation is possible in any position. Cuts dressed lumber to a maximum depth of 3 in.; cuts $2\frac{1}{8}$ in. deep at a 45-degree bevel. Universal fan-cooled motor operates at 4500 rpm. Saw weighs $13\frac{1}{2}$ lb. *Porter-Cable Machine Co., Syracuse, N. Y.*

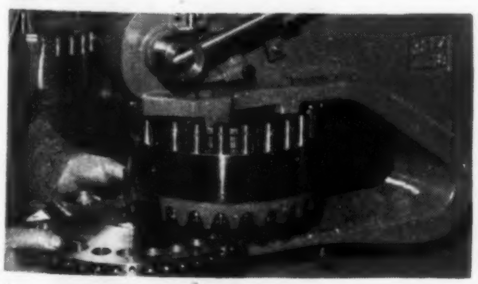
ROTEX PUNCH PRESSES Have SPEED CASE Free Machining Plate Steel Die Turret Plate



Speed Case (X1515) low carbon, open hearth, steel plate has won the favor of numerous manufacturers of machine tools, because of its machinability, its ease of finishing and response to the various case hardening methods.

Typical is Rotex Punch Co., Oakland, California, who adopted **Speed Case** for the die turret plate on their versatile presses. They report ease of machining, longer tool life and negligible distortion in heat treating.

The die turret plate retains the dies which are readily removed and replaced. At a flick of the wrist a different size punch can be brought into position. The turret die plate is then rotated to desired position and locked in place. Rotex "Quick Change" Punch Presses are widely used in sheet metal, fiberboard, plastic and other fabricating plants. Punches and dies may be round, square or other shapes. **Speed Case** has ideal physical properties for this application. Perhaps it can save time and cut costs on your equipment.



SEND FOR THE FACTS!

DISTRIBUTED BY

Brown-Wales Co., Cambridge-Hartford-Auburn — Bridgeport Steel Co., Milford, Conn. — Beals, McCarthy & Rogers, Buffalo, N. Y. — Burger Iron Co., Akron, Ohio — Grammer, Dempsey & Hudson, Inc., Newark, N. J. — Earle M. Jorgensen Co., Los Angeles-Houston-Oakland-Dallas — Passaic County Steel Service, Inc., Paterson, N. J. — Peckover's Ltd., Montreal-Toronto — Peninsular Steel Co., Detroit, Mich. — Pidgeon-Thomas Iron Co., Memphis, Tenn. — Horace T. Potts Co., Philadelphia-Baltimore-York, Penna.

Produced by W. J. Holliday & Co., Inc., Speed Steel Plate Division, Hammond, Indiana. Plants: Hammond and Indianapolis, Indiana.

Speed Steels

SINCE 1856

W. J. HOLLIDAY & CO. (INC.)

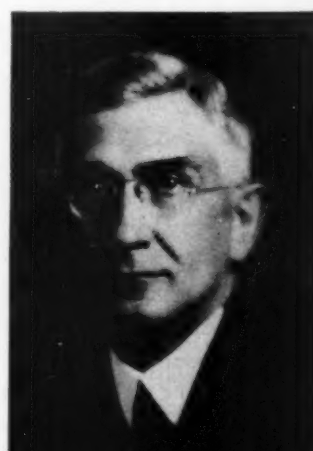
SPEED ALLOY • SPEED CASE • SPEED TREAT

MEN
OF MACHINES

H. H. Gorrie

H. H. Gorrie, formerly assistant chief engineer of the Bailey Meter Co., Cleveland, has been appointed chief engineer, to be directly in charge of the company's engineering activities. Mr. Gorrie, a graduate of Rensselaer Polytechnic Institute, joined Bailey Meter in 1927 and was appointed assistant chief engineer in 1944. He is a registered professional engineer in Ohio and is active in the American Society of Mechanical Engineers, the American Society of Testing Materials, the Scientific Apparatus Makers Association and the Cleveland Engineering Society.

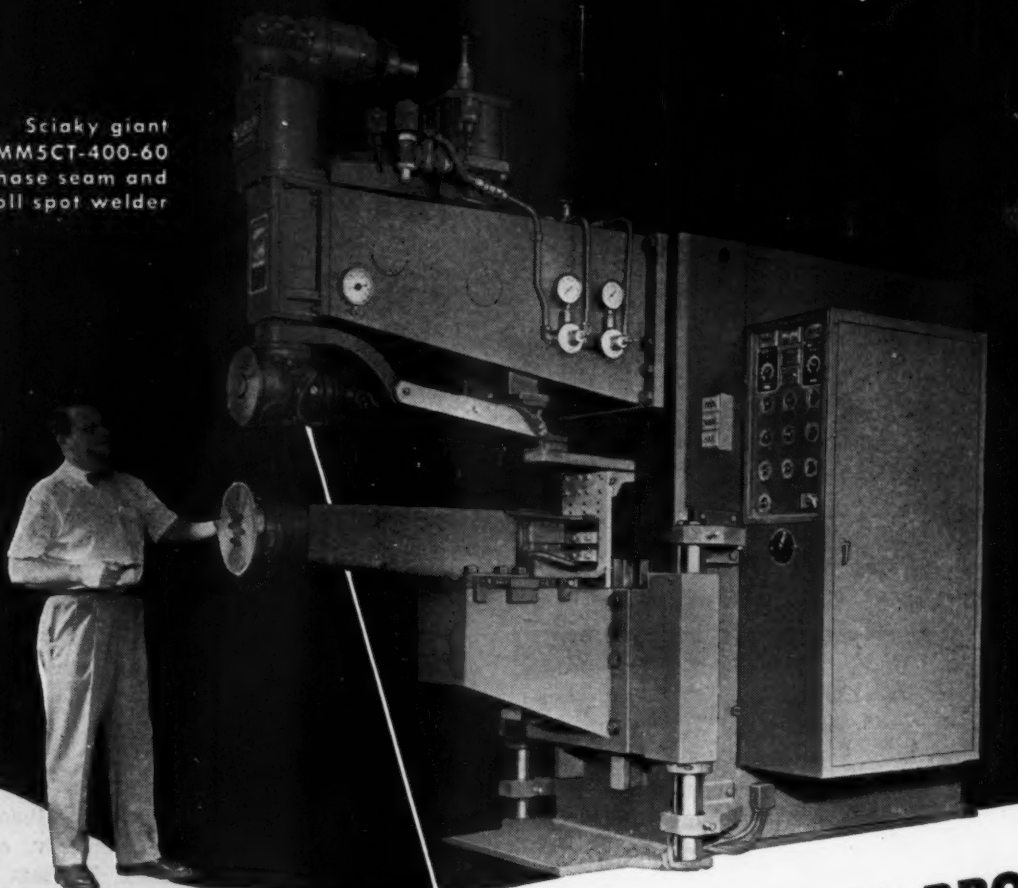
♦



Lewis K. Sillcox

The nomination of **Lewis K. Sillcox** for 1954 president of the American Society of Mechanical Engineers has been announced by the society. He will begin his term of office at the conclusion of the ASME annual meeting in November. Mr. Sillcox was educated at Trinity School, New York, and L'Ecole Polytechnique, Brussels. He began his career in the transportation industry in 1903 as a railroad foundry apprentice. From 1918 to 1927 he was general superintendent of motive power of the Chicago, Milwaukee & St. Paul Railroad, and in the latter year became first vice president of the New York Air Brake Co. In 1948 he was made a director, the following year was elected executive vice president, and in 1952 was elected vice-chairman of the board, the position he now holds. During the past

Sciaky giant
PMM5CT-400-60
three phase seam and
roll spot welder



Where ordinary relays won't do...
...that's where you find
CLARE RELAYS

IN SCIAKY BROS. GIANT SEAM WELDERS, for instance

CLARE
Type C Relay

CLARE
Type J Relay

Clare Relays which perform precise functions in control of Sciaky welders are of plug-in type with dust covers

There is nothing small . . . or ordinary . . . about this giant SCIAKY SEAM WELDER. It has a 54" throat depth for longitudinal seam welding . . . a 60" throat depth for circular seam welding . . . and will deliver a maximum of 150,000 amperes of welding current between the electrodes.

Engineers of Sciaky Bros., internationally famous pioneers and inventors of electric resistance welding equipment, found CLARE RELAYS a natural selection for their controls which demand the utmost in precise operation. Among the requirements were:

★ **Extremely fast operation . . . Within half a cycle, two relays — one operating another — have to be energized and operate their contacts.**

★ **Complete structural uniformity . . . Two relays with coils in series must operate their contacts simultaneously and maintain this performance consistently.**

No, this was no job for ordinary relays. Clare Relays were selected and are used for important precision controls in all Sciaky control panels.

If your product, like the Sciaky welder, is a quality product . . . if you must have only the highest quality components . . . a consultation with the engineers of C. P. Clare & Co. can save you time and money. Experienced sales engineers are located in principal cities. Call them direct or write: C. P. Clare & Co., 4719 West Sunnyside Avenue, Chicago 30, Illinois. In Canada: Canadian Line Materials Ltd., Toronto 13. Cable address: CLARELAY.

FIRST IN THE INDUSTRIAL FIELD

CLARE RELAYS

ten years he has been responsible for developing a brake cylinder release valve for freight cars, improved sanding methods for rail conditioning, a brake pipe feed valve and basic brake types. He also designed several types of aircraft hydraulic pumps. A Fellow of the ASME, to which he has belonged since 1916, Mr. Sillcox was made an Honorary Member in 1946. In 1943 he was awarded the ASME Medal for distinguished service in engineering and science.

◆

At the Summer General Meeting of the American Institute of Electrical Engineers, **Elgin B. Robertson** was elected president of the institute for 1953-1954. Mr. Robertson is president of Elgin B. Robertson Inc., Dallas, Tex.

◆

K. V. Hackman is now president of Southwest Products Co., Duarte, Calif. Mr. Hackman also serves as chief engineer of the company.

◆

Recently appointed manager of technical operations of the Aviation Gas Turbine Div. of Westinghouse Electric Corp., Philadelphia, **Thomas A. Daly** will be responsible for the co-ordination and overall direction of technical operations at both the Philadelphia and Kansas City, Mo. jet plants. Mr. Daly joined the company in 1942.

◆

J. A. Wilson Jr. has been appointed assistant chief engineer of the C. A. Norgren Co., Englewood, Colo. Mr. Wilson has been associated with the company since 1950 as chief draftsman and then as supervisor of design and drafting. He also served in engineering capacities at Boeing Aircraft, Winter Weiss Co. and Silver Engineering Works Inc. before joining Norgren.

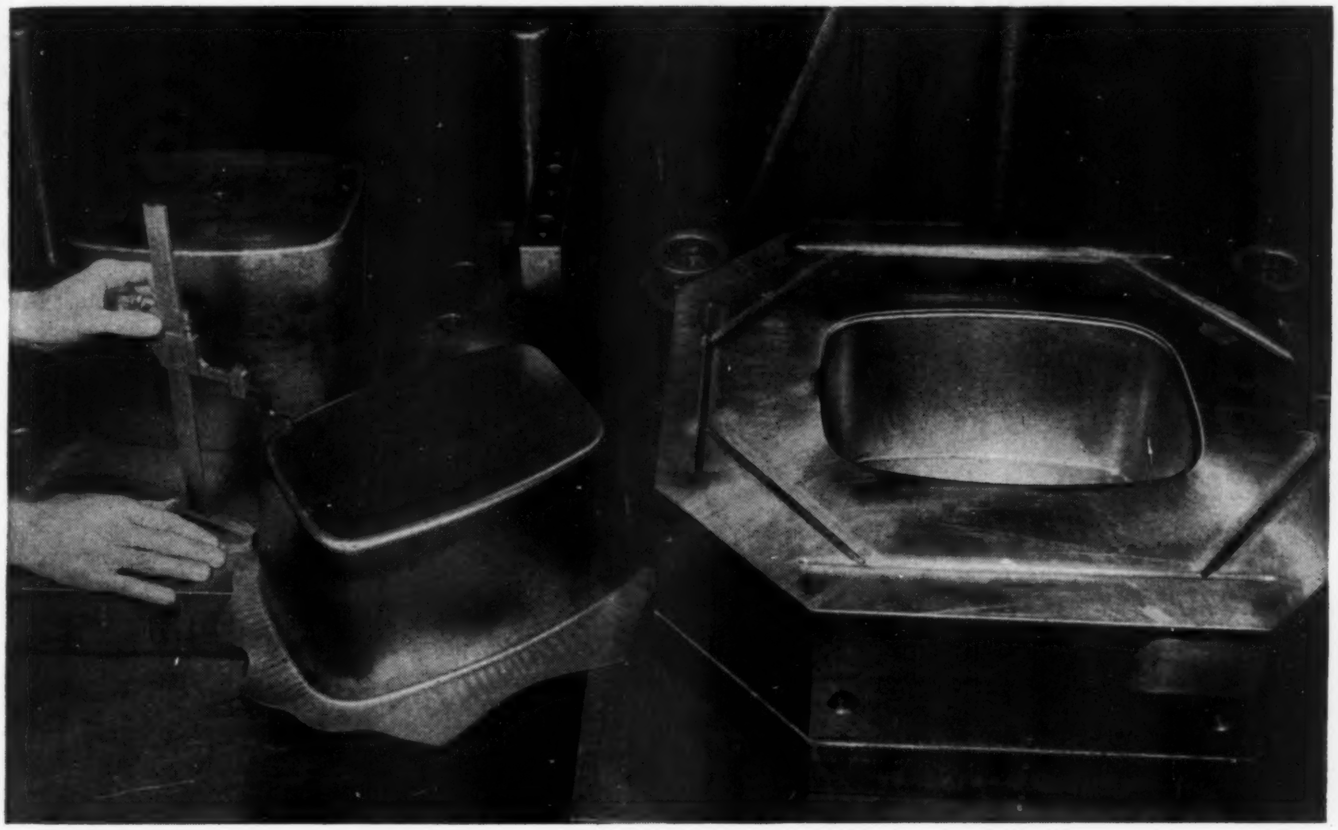
◆

Recently promoted to chief engineer of Potter & Johnston Co., Pawtucket, R. I., **Gordon W. Smithson** will direct all product improvement, engineering and research, both mechanical and electrical, on the company's line of machine tools.

◆

Elgin National Watch Co., Elgin, Ill., has appointed **C. N. Challacombe** as assistant director of research. **Earl H. Schaefer** has been promoted from chief manufacturing engineer to head of the entire engineering department; **Max Favret** is now technical co-ordinator for the company; and **Richard W. McCornack** is chief production engineer. Dr. Challacombe joined the company in 1942 as a senior research engineer and was later promoted to chief product development engineer. Mr. Schaefer has served as research engineer, assembling department head and assistant manager of the company's Lincoln, Neb., factory since joining the company in 1941. Mr. Favret came to Elgin in 1949 from Switzerland where he was chief engineer of the Langendorf Watch Factory and as technical and man-

Graph-Mo® steel die draws contour shell 5⅞" deep without scoring!



TO keep production up and finishing costs down when deep drawing contour shells for home fryers, the Dickey Grabler Company, Cleveland, Ohio, uses dies made of Graph-Mo®—one of four Timken® graphitic tool steels.

Because of the free graphite in its structure, Graph-Mo has outstanding low-friction properties: minimum tendency to score, seize, scuff or gall. As a result, Graph-Mo dies make the difficult 5⅞" deep contour draw without scoring the stamping. Costly polishing before the chrome-plating operation is eliminated, die life increased.

Graph-Mo offers other die advantages, too. The diamond-hard carbides in Graph-Mo give excellent resistance to wear and abrasion. It responds uniformly to heat-treatment, permitting closer tolerances in the parts. It has good stability, and is easier to machine than ordinary tool steels.

For more helpful information on the four Timken graphitic tool steels and their uses in dies, punches, gages and machine parts, write for the 10th edition of "Timken Graphitic Steel Data Book". The Timken Roller Bearing Company, Steel and Tube Division, Canton 6, Ohio. Cable address: "TIMROSCO".

YEARS AHEAD—THROUGH EXPERIENCE AND RESEARCH

SPECIALISTS IN FINE ALLOY STEELS, GRAPHITIC TOOL STEELS AND SEAMLESS TUBING

ufacturing director of watchmaking at the O. Schild watch factory. Mr. McCornack has been a member of Elgin's production engineering department since 1946.

Chase Donaldson has been elected president of East Coast Aeronautics Inc., Pelham Manor, N. Y., a subsidiary of Barium Steel Corp. Since 1946 Mr. Donaldson has served as consulting engineer on new products and processes for several industrial companies.

New assistant to the president of Doerr Electric Corp., Cedarburg, Wis., is **Karl A. Blind,** whose duties will be mainly the development of new products. Previous to joining Doerr, Mr. Blind was chief engineer of the Dings Magnetic Separator Co.

L. Eugene Root has been appointed director of development planning by Lockheed Aircraft Corp., Burbank, Calif., to head a new program of aviation development. Under the direction of Mr. Root, a group of engineering scientists will seek to forecast aviation trends 10 to 25 years in the future. Endeavoring to hasten future developments, the development planning group will study directions of aviation development; trends in power, speed and aircraft capacity; utilization of accumulated scientific knowledge; and will forecast generally corporate activities in future weapons systems involving new types of airplanes and missiles. Mr. Root holds master's degrees in mechanical and aeronautical engineering from California Institute of Technology and has had 19 years of varied experience in aviation.

The Pannier Corp., Pittsburgh, has appointed **J. M. Hendrickson** to the position of vice president in charge of research and development. He was formerly director of mechanical research at the Eljer Co., in charge of product design and development.

Louis H. Schuette, vice president in charge of the Hydraulic Div. of Sundstrand Machine Tool Co., Rockford, Ill., has been elected a director.

Former research engineer at the Borg-Warner central research laboratory, **Donald H. Madsen** has been named a research engineer in the heat-power department of Armour Research Foundation of Illinois Institute of Technology.

Benjamin Cametti has been appointed manager of the pump engineering section of the Atomic Equipment Department of the Westinghouse Electric Corp., Pittsburgh. He joined the company in 1925 and has served in various capacities, including tool designer and engineering draftsman. He was chief draftsman at the research laboratories from 1934 to 1942 and a research engineer from 1942 to 1949.

THE ENGINEER'S Library

Recent Books

Applied Kinematics. *By J. Harland Billings, professor of mechanical engineering, Drexel Institute of Technology; 360 pages, 6 by 9 inches, clothbound; published by D. Van Nostrand Co. Inc., New York; available from* MACHINE DESIGN, *$4.50 postpaid.*

A thorough solution of problems involved in the design of machines and their mechanisms requires a comprehensive understanding of kinematics. This textbook fills this need and a study of it would provide adequate preparation for work on servomechanisms and analog computers. About the first half of the book is comprised of the basic subjects of motion in machines, various kinds of mechanisms including cams, and a discussion of velocities and accelerations. A rather large section is devoted to gears and gearing while the subjects of automatic control and computers are treated in the final chapter.

•

Electric Control Systems. *By Richard W. Jones, professor of electrical engineering, Northwestern Technological Institute; 525 pages, 6 by 9¼ inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, *$7.75 postpaid.*

Written with the assumption that the reader has had the normal background in circuit theory, electric machinery, electronics, and differential equations, this book is concerned primarily with control systems for electric motor drives. Dc and ac motor characteristics are thoroughly covered in the first sections of the book followed by a study of switching and contact-making devices and circuits. In succeeding chapters a treatment is made of power amplifiers, control circuits, motor acceleration and speed control, motor braking and protection, and feed back control.

•

Materials Handling. *By John R. Immer, professor of industrial management, American University; 599 pages, 6¼ by 9¼ inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, *$8.00 postpaid.*

This text is the product of the development many companies have made in establishing techniques, methods, and philosophies of materials handling. Consisting substantially of case histories, principles are demonstrated to show how more efficient methods of material movement can be accomplished. The subjects of time and motion study, materials handling

equipment, packaging, handling problems analysis, and organizational programs are encompassed in this volume. Chapters on operator training and instruction, maintenance, and safety are included.

♦

The High-Speed Internal-Combustion Engine. *By Harry R. Ricardo, 432 pages, 6½ by 9½ inches, clothbound; published by Blackie & Son Ltd., London; available from Autobooks, 2708 Magnolia Blvd., Burbank, Calif.;* $11.00.

A fourth edition, this book was first published in 1923. Completely rewritten, this edition concentrates on research, design and development work in this field with particular emphasis on the subject of mechanical efficiency. Thorough detailed cross-sectional drawings of internal combustion engines are found throughout the volume.

♦

Vacuum Tube Oscillators. *By William A. Edson, Stanford University; 492 pages, 6 by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, *$7.50 postpaid.*

The transient behavior of linear systems, negative resistance oscillators, nonlinear oscillations, feedback systems and stability criteria, and resonators are introduced in the first chapters of this book. A coverage is made of linear, harmonic, crystal-controlled, and relaxation oscillators. Other topics dealt with include locking and synchronization, frequency multiplication and division, modulation of oscillators, and automatic frequency control.

Manufacturers' Publications

Simple Blueprint Reading with Special Reference to Welding and Welding Symbols. 207 *pages, 5½ by 8½ inches, clothbound; available from Lincoln Electric Co., 22801 St. Clair, Cleveland, O.* $1.00.

A fourth edition, this book contains new information, more drawings, and photographs along with a complete revision of symbols and other welding data to conform to the latest standards. The material is intended to provide a clear understanding of the use of blueprints in mechanical fabrication and construction. After basic welding principles are explained, welding symbols and their use are fully described.

♦

Welding Aluminum. 186 *pages, 6 by 9 inches, paperbound; available from Reynolds Metals Co., 2500 S. Third St., Louisville, Ky., on company letterhead request.*

Information concerning welding, brazing, and soldering aluminum are included in this handbook reference. Gas, arc, flow, stud, induction, resistance, upset, and pressure welding are covered along with

torch, furnace, salt bath, induction and resistance brazing. Many tables and illustrations are provided throughout to augment the text.

•

Engineering Data on Thread and Form Rolling. 41 *pages,* 8½ *by* 11 *inches, paperbound; available from Reed Rolled Thread Die Co., Worcester* 1, *Mass.,* $1.00 *per copy.*

The thread rolling process, advantages and applications of rolled threads, and equipment and tools for producing rolled threads are covered in the first half of this pamphlet. In the second half rollability of materials is discussed and tables concerning rolled thread finishes and relative die life are presented. Preferred forms for rolling, blank specifications, and pertinent reference tables are also included.

Association Publications

Boron Steel. 116 *pages,* 8½ *by* 11 *inches, paperbound; available from American Society for Metals,* 7301 *Euclid Ave., Cleveland* 3, *O.;* $1.00 *per copy.*

A second revised edition, this booklet is made up of panel discussions, *ASM Transactions* articles, and articles that have appeared in *Metal Progress* concerning the subject of boron steel. A section covers experiences with boron steels followed by two articles which discuss the effect of boron on steel. A hardenability test, H-steels, and their use are included in a supplement at the end.

•

American Management Association Series. *Each publication is* 6 *by* 9 *inches, paperbound; copies available from American Management Association,* 330 *W.* 42*nd St., New York* 36, *N. Y.;* $1.00 *each for members,* $1.25 *for nonmembers.*

The following publications are available:

Manufacturing Series

209. **Guides to Meeting Tomorrow's Production Needs**—64 pages.
210. **Industry Enters the Atomic Age**—31 pages.

New Standards

Safety Code for Mechanical Power-Transmission Apparatus. *ASA B*15.1-1953; 52 *pages,* 5¼ *by* 7½ *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.;* $1.00 *per copy.*

The scope of this code applies to all moving parts of equipment used in the mechanical transmission of power, including prime movers, intermediate equipment and driven machines, excluding point of operation. Safeguarding of connecting rods, cranks, flywheels, shafting, pulleys, drives, gears, sprockets, projections, cams, clutches, counterweights, revolving or

reciprocating parts, etc. are among the items covered.

♦

1952 Book of ASTM Standards. *Seven volumes, 6¼ by 9¼ inches, clothbound; available from the American Society for Testing Materials,* 1916 *Race St., Philadelphia* 3 *Pa.; complete set*—$76.00; *parts* 1, 3, 6—$12.00 *each; parts* 2, 4, 5, 7—$10.00 *each.*

More than 2000 standards, specifications, tests, and definitions for materials are compiled in this 9975-page book which is made up in seven parts. The parts comprising the complete set are:

Part 1—Ferrous Metals; 291 standards, 1602 pages.
Part 2—Nonferrous Metals; 268 standards, 1359 pages.
Part 3—Cement, Concrete, Ceramics, Thermal Insulation, Road Materials, Waterproofing and Soils; 405 standards, 1666 pages.
Part 4—Paint, Naval Stores, Woods, Fire Tests, Sandwich Constructions, Building Constructions; 273 standards, 1182 pages.
Part 5—Fuels, Petroleum, Aromatic Hydrocarbons, and Engine Antifreezes; 213 standards, 1282 pages.
Part 6—Rubber, Plastics and Electrical Insulation; 257 standards, 1520 pages.
Part 7—Textiles, Soap, Water, Paper, Adhesives and Shipping Containers; 294 standards, 1364 pages.

Each part has a detailed subject index and two tables of contents. A complete index to standards is also furnished without additional charge. To keep the material up-to-date, a supplement to each part will be issued in 1953.

♦

Spindle Noses and Arbors for Milling Machines. *ASA B5.18-1953;* 7 *pages,* 8½ *by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W. 39th St., New York* 18, *N. Y.;* $1.00 *per copy.*

Drawings and essential dimensions of spindle noses, arbor and adapter ends, and draw-in bolt ends are covered. All dimensions are given in inches, and sizes 30, 40, 50 and 60 make up the standard.

Government Publications

NACA Technical Series. *Each publication is* 8 *by* 10½ *inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics,* 1924 *F St. N.W., Washington* 25, *D. C.*

The following Technical Notes are available:

2924. **Combined Stress Fatigue Strength of 76 S-T61 Aluminum Alloy with Superimposed Mean Stresses and Corrections for Yielding**—90 pages.

2930. **Strength Analysis of Stiffened Thick Beam Webs with Ratios of Web Depth to Web Thickness of Approximately 60**—11 pages.

2939. **Optimum Controllers for Linear Closed-Loop Systems**—27 pages.

2940. **Effect of High Bulk Temperatures of Boundary Lubrication of Steel Surfaces by Synthetic Fluids**—27 pages.

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Making "picture packages" for Lipton's

Have you noticed the new trend in food packaging? Today, every food producer wants color in his packages—bright, rich color that will catch the customer's eye—and hold it!

One thing that helps to make these "picture packages" possible is a new method of lamination used to hold a transparent film (on which the colorful advertising matter is printed) to a protective foil (in which the food will be contained).

Many packagers rely on 3M adhesives to bond film and foil together. Some, for example, use a 3M adhesive which is completely transparent when dry. What's more, it forms an immediate bond once applied. And its fast-drying property makes it possible to laminate packages on machines running at high speeds.

See what adhesives can do for you . . .

Making "picture packages" for Lipton's is another example of how 3M helps industry solve problems. For ideas on how 3M can help you, call in your 3M sales representative, or write for free, booklet describing many applications. Address: 3M, Dept. 198, 411 Piquette Ave., Detroit 2, Mich.

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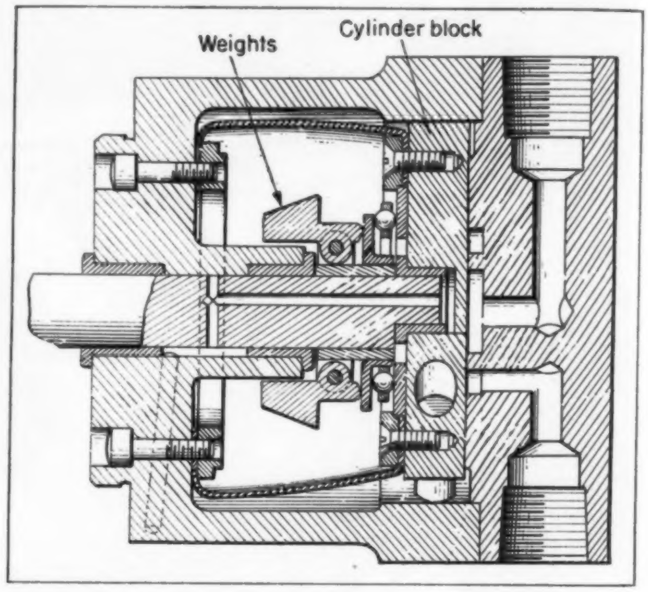
NOTEWORTHY Patents

INSTANTANEOUS POWER REVERSAL for fractional horsepower motors is achieved in a motor system designed by Jacob Rabinow and detailed in patent 2,632,141. Assigned to the United States of America as represented by the Secretary of the Army, the system employs a long torsion bar fastened to one end of a conventional motor with a double-ended shaft. In normal operation, the motor is free to rotate in either direction. However, actuation of a switch to reverse the motor rotation simultaneously operates a toothed-clutch arrangement to lock the torsion bar at its far end. Rotational energy of the motor is absorbed instantly by the "winding up" of the bar, which also acts as a torsion spring to throw the motor in reverse. Designed for computer applications, the system has been used successfully to reverse a 1/20-horsepower motor in 1/200-second.

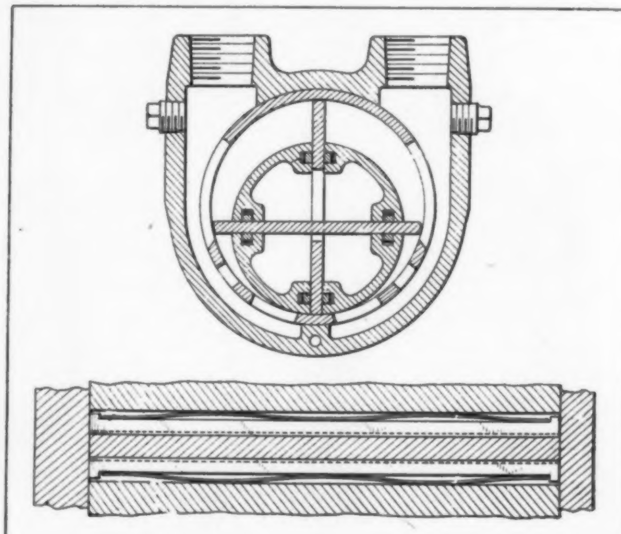
MINIMUM STARTING DEMANDS reduce initial power requirements in a radial piston type hydraulic pump and permit rapid acceleration to full load conditions. Designed by Floyd E. Carlson, the pump uses a clever flyball governor and cam mechanism to control the output pressure of the pump. Pumping action is provided by an eccentric on the drive shaft which imparts a circular motion to a nonrotatable cylinder



block, driving radial pistons carried in the block. Sealing pressure between the block and an end wall containing the fluid ports is varied by the flyball governor

device so that full pumping pressure is not developed until the shaft reaches approximately full speed. Braking of the load after power cutoff is gradual, permitting the motor and pump to coast to a stop. The pump design is covered in patent 2,621,606, which has been assigned to Sundstrand Machine Tool Co.

LOW-FRICTION SEALING of reciprocating vanes in a rotary impeller pump for high-pressure pneumatic operation minimizes leakage and obviates close-tolerance machining requirements of conventional types. In the design covered in patent 2,625,112, flat vanes, sliding back and forth in slots in a cylindrical rotor which is offset to provide pumping action, are sealed

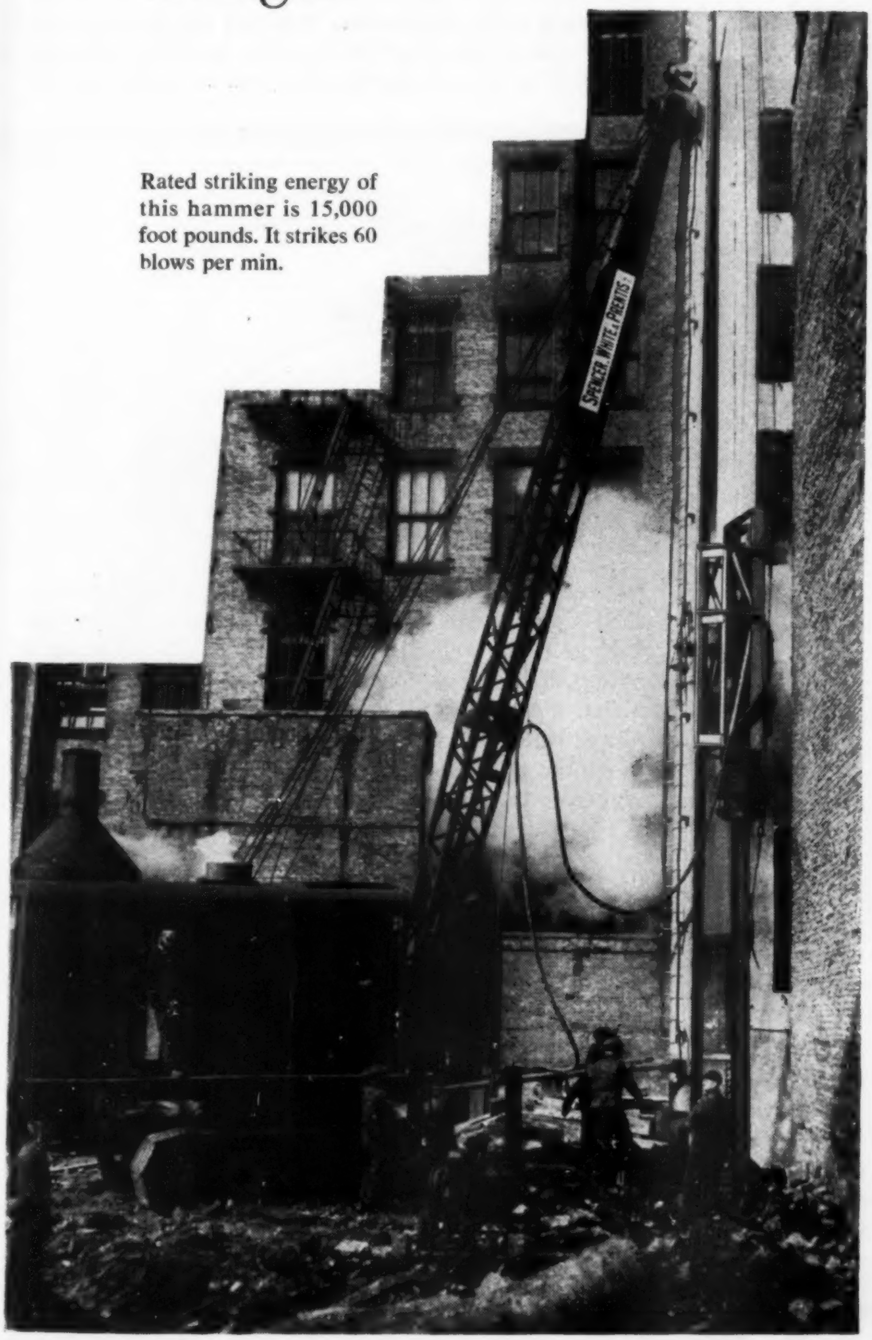


by spring-loaded lead-bronze shoes mounted adjacent to the rotor slots. Dimensional changes in the slots and vanes due to pressure and temperature variations at high speeds which would have an adverse leakage effect are compensated automatically and the centering action of the shoes reduces friction and wear at the slots. Actual tests have shown substantial improvements in efficiency accompanied by reduced temperatures of operation. Inventor Carl J. Stubau has assigned the patent to Borg-Warner Corp.

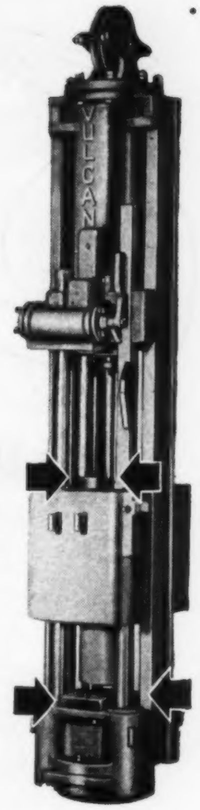
EXCESSIVE PRESSURE PROTECTION for hydraulic systems is afforded by a relief valve assigned to the New York Air Brake Co. Designed by William T. Stephens, the valve covered in patent 2,630,825 is employed between the high pressure side of a pump and a reservoir. In normal operation, flow bypasses the valve and goes directly to the load. However, excessive pressure buildup in the lines actuates a cylindrical spring-loaded plunger in the valve permitting fluid to be discharged to the reservoir until the pressure conditions return to normal. Simple and com-

"U. S." cushions a pile hammer ...using a 53-year-old design

Rated striking energy of this hammer is 15,000 foot pounds. It strikes 60 blows per min.



Arrows indicate the "U. S." rubber bumpers. Despite the terrific series of impacts, they retain their natural resilience.



This pile hammer is equipped with 2 "U. S." rubber bumpers at the top of the ram, and 2 at the base. It's their job to absorb the smashing over-drive of the hammer and prolong the life of the equipment. But the main point is that for the last *53 years these bumpers have been built according to the same blueprint*. So perfect was the design that the makers of the hammer have never required any change.

This is an example of the quality built into every United States Rubber Company product. "U. S." engineers stand ready to work with you in the improvement of any product. Consult them at any of our 25 District Sales Offices. Or write to address below.

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Another "U. S." leader that has rolled up enviable records for long service is Matchless® High Pressure Steam and Pile Driver Hose. It is burst-proof, for the protection of the worker.

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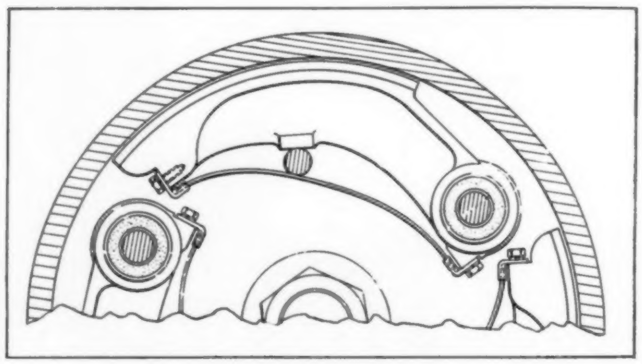
pact, the valve construction eliminates chattering during operation, permits handling of large flow volumes and facilitates assembly and maintenance. Several modifications for adapting the valve to varying pressure and temperature conditions are also described.

AUTOMATIC FRICTION ADJUSTMENT, proportional to speed, is obtained with "floating" flat springs in a centrifugal clutch detailed in patent 2,626,034. Clutching action is provided by shoes faced with brake lining and pivoted at one end in resilient bushings mounted to the driving member, which swing out un-



der rotation to engage a drum on the driven member. Centrifugal force of engagement is controlled by flat springs freely mounted across the back of the shoes in stamped fittings which abut and are restrained by pin stops. Assuring positive engagement and release, clutch construction also minimizes wear and facilitates assembly and maintenance. The patent has been assigned to the Fawick Airflex Co. by Thomas L. Fawick.

PRECISION FLOW CONTROL of small fluid amounts at low pressures is afforded by a novel rotary pump designed by Roger G. Olden. Described in patent 2,629,333, the pump consists of a heavy wire rotor wound in the shape of a helix which constricts, at its outside diameter, one or more hoses mounted parallel to the helix axis. Rotation of the rotor about its own axis progressively constricts each hose to provide a pumping and suction action. Uniform flow is assured and the quantity pumped is determined by the pitch of the helix and the rotor speed. Compact in construction, the pump may be used to deliver several different liquids simultaneously by varying the arrangement of hose sections. Assignee of the patent is the United States of America as represented by the Atomic Energy Commission.

Copies of all patents presented in this department may be obtained for 25 cents each from The Commissioner of Patents, Washington 25, D. C.